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FRICTION CLUTCHES

A HANDBOOK FOR
MECHANICAL ENGINEERS, AUTOMOBILE
ENGINEERS, DESIGNERS AND STUDENTS

BY

R. WARING-BROWN

A.M.I.A.E., F.R.S.A., M.I.P.E.

WITH 75 ILLUSTRATIONS

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PREFACE

THE purpose of the author in preparing this primer has been to present in concise and complete form the fundamental principles involved in the design and operation of friction clutches.

The work is primarily intended for students, and it is also hoped that it may prove of service to mechanical engineers, automobile engineers, and designers.

Recent years have produced many new and interesting designs, while development has been greatly enhanced by the introduction of new friction materials, descriptions of which may not be found in standard text-books.

Hence many recent devices are illustrated and described, while modern applications of friction clutches to many types of machinery are shown.

In the preparation of the manuscript the author has consulted many standard works and the technical press, and wherever information from such sources has been used, has endeavoured to give suitable acknowledgment.

The numerous illustrations shown have been selected as the most representative of current practice.

Through the generosity of various manufacturers at home and abroad, modern examples of friction clutch design have been placed at the disposal of the writer, who sincerely acknowledges his indebtedness.

R. WARING-BROWN.

22 QUEEN VICTORIA ROAD,
COVENTRY.

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FRICTION CLUTCHES

CHAPTER I

FUNCTIONS OF FRICTION CLUTCHES

General. The useful application of friction is embodied in clutch mechanisms for the transmission of motion.

Friction clutches are built in many different designs, with many modifications, which combine the features of various types.

The class of friction clutch to adopt is generally decided by the specific purpose for which it is to be utilized. For instance, in clutches for automobile transmissions the matter of absolute disengagement is very important, while weight is a great factor, especially as affecting inertia, because a clutch having a high flywheel effect spins to such an extent as to cause clashing of gears.

Again, in hoisting machinery it is important that certain motions be engaged without stopping the motor, and also that the two portions of the clutch should be locked in frictional engagement.

Further, a particular type of clutch is required where it is necessary to start and accelerate a load requiring a large starting torque.

Machine tools necessitate the use of clutches which take up a small amount of space, and yet which are capable of withstanding rough usage.

Motor cycles demand unusually small friction clutches of large power-transmitting capacity, permitting high pressures between the friction surfaces.

All these, and many others, have their own particular requirements, to be met by special design.

Classification. Friction clutches may be grouped into

two general classes, determined by the direction in which the pressure is applied between the contact surfaces, as follows—

1. Axial Clutches. Which include all those clutches having the contact pressure applied in a direction parallel to the axis of rotation, as in all types of cone clutches, disc or plate clutches, or combination of both types.

2. Rim Clutches. Which include all clutches wherein the contact pressure is applied upon a rim or drum, in a direction at right angles to the axis of rotation.

Rim clutches may be further divided into—block, band, split ring and roller types.

Requirements of a Friction Clutch. The primary object of a friction clutch is to connect a rotating member to one that is stationary in such a manner that inertia will be smoothly overcome, and the subsequent motion accelerated to required speed with a minimum stress on the parts of the mechanism, when the full power of the prime mover must be transmitted with the minimum amount of slip.

In designing a clutch the following points must be given careful consideration—

1. The materials forming the contact surfaces must be selected with care.

2. Sufficient gripping power must be provided, so that the load may be transmitted in the shortest possible time.

3. In order to keep the inertia as low as possible, a clutch should be made as light as conditions render advisable. This is most important in high speed service, such as is usual in automobiles.

4. A clutch should be simple in design, and should contain as few parts as possible.

5. The motion should be transmitted without shock or involuntary slipping.

6. A clutch should have the ability to slip under load for considerable periods, without unduly heating up.

7. Provision should be made for dissipating the heat generated at the contact surfaces.

8. Provision should also be made for instantaneous disconnection between the prime mover and the transmission, without drag.

9. A clutch intended for high speed service must be carefully balanced.

10. A clutch should be self-contained, and should have as few projecting parts as possible.

11. The driving and driven members forming the clutch should be automatically capable of self-alignment.

12. The construction should be such as to facilitate repair and adjustment.

CHAPTER II

FRICTION MATERIALS

Friction. The motion which takes place when one body is moved upon another is resisted by "friction," and may be defined as "that force which acts between two bodies at their surface of contact, so as to resist their sliding on each other."

The force of friction F bears a certain relation to the pressure between the two bodies, according to the conditions under which sliding occurs. This pressure is called the normal pressure N.

The relation between force of friction and normal pressure is given by the coefficient of friction denoted by the Greek letter ; thus—

$$F = \mu \times N \text{ and } \mu = \frac{F}{N}$$

If a body is placed on an inclined plane, the friction between the body and the plane will prevent it sliding down the inclined surface, provided the tilt of the plane is not too great. There will be a certain angle at which the body will just barely remain stationary, the frictional resistance being nearly overcome by the tendency of the body to slide down.

This angle is termed the "angle of repose," the tangent of which is equal to the coefficient of friction. This angle is usually denoted by the Greek letter ; thus—

$$\mu = tan\ \theta$$

If two bodies are pressed together so that unital pressure perpendicular to the contact surfaces is N, and F is force of friction to be overcome before sliding will take place, then $\frac{F}{N}$ is called the STATIC COEFFICIENT OF FRICTION.

If the two bodies are sliding steadily on each other

and F^1 is the steady resistance to sliding, then $\frac{F^1}{N}$ is termed the KINETIC COEFFICIENT OF FRICTION.

A greater force is required to start a body from rest than to keep it in motion, because the friction of rest, or static friction, is greater than the friction of motion, or kinetic friction.

Laws of Friction for Dry Surfaces. The following may be considered as a reasonable statement of the laws of friction for dry metallic surfaces—

1. For low pressure the friction is directly proportional to the normal pressure between the contact surfaces. As the pressure increases to a high value the friction does not rise in proportion. When the pressure becomes abnormally high, the friction increases rapidly until seizing takes place.

2. The friction in its total amount and its coefficient is independent of the areas in contact, so long as the total pressure remains constant, where only moderate pressures are involved. For high pressures this law is modified, as in the first case.

3. At very low velocities the friction is independent of the rubbing velocities, but as the velocities increase the friction decreases.

Laws of Friction for Lubricated Surfaces. The laws for lubricated surfaces differ considerably from those governing dry surfaces.

1. The frictional resistance is almost independent of the unit pressure if the surfaces are flooded in oil.

2. The frictional resistance depends largely upon the temperature, because of the change in the viscosity of the lubricant.

3. The friction varies as the speed at low unit pressures, but for high pressures the friction is very great at low velocities.

4. If the friction surfaces are immersed in oil, the friction is independent of the nature of the materials. With the reduction of lubricant the friction becomes more dependent upon the materials forming the contact surfaces.

Lubrication. The question of lubrication in relation to

friction surfaces of clutches is only of vital importance wherein such surfaces are of the multiple or plate metal to metal types.

The value of an oil as a lubricant depends to a great extent on the viscosity of the oil, although this does not necessarily mean that an oil of the highest viscosity is the most suitable lubricant.

Again, an oil of the lowest viscosity, which will retain an unbroken oil film between the friction surfaces, is the most suitable for lubrication purposes, because a higher viscosity than that necessary to retain an oil film results in a waste of power, due to the expenditure of energy necessary to overcome the internal friction of the oil itself.

It is considered that vegetable or animal oils are unsuitable, owing to the fact that they decompose at high temperatures and frequently have a tendency to corrode metals with which they are in contact.

The requirements of metal to metal clutches are very severe. The surfaces should be covered with lubricant when there is any slippage whatever ; at the same time it is desirable that when the maximum pressure is applied the lubricant should be forced away from the surfaces, to enable the clutch to take up the drive, which will be better if the surfaces are dry.

One defect of the metal to metal clutch is its tendency to "drag," which effect can be considerably increased if the lubricant is unsuitable in quality or quantity.

A thin oil will enable the clutch to operate efficiently, while a more viscous lubricant will cushion its action. Further the lubricant may be subjected to considerable heat due to slippage, and must have the ability to withstand this without denaturing.

Coefficients of Friction. Experiments prove that there is considerable difference between the static and kinetic coefficients of friction. The peculiar characteristics of these friction coefficients are conclusively shown in the useful application of friction to clutches for power transmission, and particularly so in reference to automobile purposes.

Consider the operation of a clutch. The initial engagement causes the driving surface to slip against the driven surface at widely different velocities, the relative difference being automatically reduced as the drive is taken up. Directly the differences of movement disappear there is an increase in driving effort.

Taking this into account, together with the questions of material, pressure, temperature and lubrication, it is evident that so many variable factors render it impossible to assign values to the coefficient of friction to meet all conditions.

The accompanying table, however, gives coefficients of friction for various materials in general use, with average figures determined by practice and experiment.

The allowable values represent approximately 50 per cent of the average values, and calculations based on these figures will be found to cover general requirements.

Heat Radiation. The friction in clutches must be distributed over a great enough area to prevent undue heating of the parts in contact. One square inch of friction surface may be allowed for 750 foot-lbs. of energy absorbed, but under ideal conditions this figure may be as much as 10,000 foot-lbs. of energy per square inch, it being largely a question of experiment.

However, to facilitate radiation of heat, ribbed exteriors are sometimes useful, while all radiating surfaces should be left rough, as polished surfaces confine the heat instead of throwing it off into the atmosphere.

Materials for Contact Surfaces. The following conditions must be fulfilled for a material to prove absolutely satisfactory as a friction surface—

1. The material must have a high coefficient of friction.
2. The material must be able to resist high temperatures, caused by slippage, due to frequent operation of the clutch.
3. Material must not be affected by moisture.
4. Material must have extreme durability.

Further, there are other questions affecting the subject of materials, such as the flexibility of the drive and

lubrication of the surfaces ; hence, taken altogether the conditions are particularly severe.

In what manner such conditions can be fulfilled will be evident from a brief description of materials found in actual practice, viz., wood, paper, leather, metals, cork and fabric.

Wood. Many kinds of wood have been used as friction surfaces for clutches applied to hoisting machinery and line-shaft transmission. Such woods as lignum-vitae, maple, elm and basswood have given satisfaction, the opposing friction surface usually being of cast iron.

These contact surfaces give a fairly high coefficient of friction, but are not satisfactory for high speeds, because they will not resist high temperature or wear.

Paper. Compressed straw-board may be used as a friction surface in clutches where the speeds and pressures are very low. Vulcanized fibre, which is really paper chemically treated, also gives fairly good service under the above conditions.

Continuous presence of oil on paper surfaces has usually a deleterious effect, while constant slippage will char the surfaces rapidly.

As will be seen from Table I, the friction coefficient is fairly high, but the material cannot be recommended other than for light drives and pressures.

Leather. This material as a friction surface has been extensively used for general machinery and automobile purposes, chiefly in connection with cone clutches.

Before leather is applied it is usually treated either by boiling in tallow, or by soaking in collan oil, castor oil, or neat's foot oil, and finally by passing through rollers to press out excess oil.

Leather facings should not be allowed to become entirely dry, or engagement of the clutch will be fierce.

There are two types of leather for this purpose—oak-tanned and chrome leathers.

Leather if subjected to constant slippage quickly chars, and then has little value as a friction material.

Metallic Surfaces. There are clutches used on line-shaft transmissions, etc., which work satisfactorily ; these are

usually of the cone type, wherein both cones are of cast iron, running dry.

Disc clutches, using hard saw-steel discs, running in oil, are advocated by some manufacturers ; others use steel against bronze, and cast iron against steel or copper.

Whatever materials of this type are used, their surfaces must be copiously lubricated with oil or graphite, or abrasion will occur and the efficiency seriously impaired.

Cork. The friction surfaces of some clutches have cork inserts, owing to the fact that cork has a very high coefficient of friction, being 0·32 when not lubricated. As a rule the cork, which has usually been softened by boiling, is pressed into suitable cavities formed in one of the metallic friction surfaces. When inserted it normally protrudes above the surrounding surface and engages first. As pressure is applied the cork, being compressible, is forced down flush with the plate, the metallic surface then acting with the cork in taking the load.

It is not absolutely essential to run these materials dry, and durability is increased by lubrication, although the friction coefficient will thereby be decreased.

Moisture has not a very detrimental effect upon cork. A typical cork insert clutch is shown in Fig. 74, the inserts covering from 25 to 50 per cent of the total area of the disc. Two forms of cork are used, one being cork in its natural condition. The other consists of small pieces of cork compressed into sheets or shapes under great pressure, and subjected to sufficient heat to cause the natural gums of the cork to exude and act as a binder. This prepared cork, while possessing great elasticity, is much stronger and more durable than the natural cork.

Fabrics. The introduction of fabric friction surfaces has had a revolutionary effect on clutch design, because results have been obtained with them that put any other kind of surface out of the running, and establish the fact that friction clutches should be worked dry and not as a compromise between a liquid lubricating the surface and the solid surface itself with the liquid squeezed or burned off.

These fabric materials were first invented by Ferodo, Ltd.,

Chapel en le Frith, who have developed them into two main varieties, viz.: Ferodo Bonded Asbestos and Ferobestos. The claims made for these materials may be summarized as follows—

1. Extraordinary high coefficient of friction.
2. Ability to withstand any temperature.
3. May be run dry or lubricated.
4. Unaffected by moisture.
5. Are non-abrasive.
6. Have extreme durability.

Ferodo Bonded Asbestos. Is composed of asbestos yarn spun on to thin brass wire, and woven into strips or belts to any desired width or thickness. After complete impregnation with certain chemical ingredients, the strip is roughly formed into a flat circle and compressed under a 1,000 ton press in dies and under heat into true annular discs. The density of the material after such pressure is nearly that of cast iron, the operation of die-pressing being a Ferodo Patent. For all annular surfaces this type is recommended, as the main strands of the woven fabric run concentrically through the body of the lining.

Ferobestos. Is a woven composite, either of woven asbestos cloth, or the natural fibres of the mineral incorporated with a soluble imporous bond. While plastic the material is forced into metal dies, and subsequently heated and compressed. By this method annular linings of any required shape or thickness can be readily produced, the faces being finished to a close degree of accuracy by surface grinding.

General Characteristics. The outstanding characteristics of these materials are their toughness, durability, high friction coefficients and their non-abrasive qualities.

The materials are porous to an extent, and will absorb a certain quantity of lubricant, without appreciable effect on the coefficient of friction.

The coefficient of friction is extremely steady under all conditions, and only shows tendency to decline at certain velocities and pressures when temperatures of between 400°–500° F. are in evidence. The coefficient claimed

for their materials is 0·38, and the results of tests carried out and given below adequately substantiate this claim—

TABLE I.

TEST OF FERODO MATERIAL.

Time. Hrs.	min.	Temperature of Rim degs. Fahr.	Coefficient of friction.	Remarks. Diagram of drum 30 in.
2	45	110°	·584	Surfaces dry. Speed constant at 1,140 ft. per minute. Specific load constant at 20 lbs. per square inch.
2	51	196°	·558	
2	52	237°	·534	
2	55	242°	·523	Speed constant at 580 ft. per min. Specific load constant at 7 lbs. per sq. inch.
4	18	70°	·647	
4	40	163°	·700	

In regard to wear, experiments show that this is almost proportional to temperature, i.e. the rate of wear is twice as great at 520° F. as at 260° F. at the similar velocities and pressures.

Unit Normal Pressures. The unit normal pressure between the contact surfaces in cone clutches is usually about 12 lbs. per square inch, although in very large sizes this may be increased to 20 lbs. per square inch. Cone clutches may work quite well under much higher pressures, according to conditions, but they naturally have lesser durability than those designed with a low unit normal pressure.

The normal pressure must be somewhat greater than the actual pressure obtained by calculation, because the axial pressure must be such as will produce the normal pressure under conditions of rest.

For instance, suppose the normal pressure is just insufficient to produce the necessary driving torque, and slippage occurs, then the pressure can only be augmented by forcing the cones into each other still further, which means that resistance has to be overcome in a direction normal to that of slippage.

It frequently happens also that the available diameter of cone is restricted in size, and as unduly wide faces are not desirable the unit normal pressure must be increased.

In metal to metal cone clutches the unit normal pressure must be much greater than necessary with leather or other friction materials to transmit the same power.

In plate clutches, which to work satisfactorily at all must be metal to metal in oil, the unit pressure varies from 0·06 to 2 lbs. per square inch, the average value being 1 lb.

In regard to fabric friction linings, the unit pressure advocated by the makers is from 8 to 18 lbs. per square inch. The writer has produced clutches with a unit pressure of 40 lbs. and more, which have proved eminently satisfactory in every way, and these materials will only wear slightly if pressures of 15 times greater than are used in average clutches are imposed.

On the question of maximum allowable pressures on friction surfaces, the point to be considered is not the possibility of wear, but the practical difficulty in releasing high pressures and the possible destructive effect of heavy end thrusts.

Friction Linings. When leather is used for lining the clutch cone, it has to be cut from a strip of the material, and on reference to Fig. 1 it is evident that it must form a sector of an annular ring. From the following formula perfect sizing of the lining may be accomplished—

Where R = outside radius of clutch leather,

r = radius of clutch cone and thickness of the material,

a = angle of the cone,

b = angle of the clutch material sector,

Then R = cosecant $a \times r$.

$$b = \frac{360^\circ}{R} \times r$$

Where fabric linings are used these can be supplied to accurate shapes, but the formula remains.

Linings may be fixed or floating, this being really only a question of design.

Floating Linings. These of course only refer to linings of the disc type. No increase in friction arises from floating linings as compared with those definitely fixed. For linings of large diameter floating types are not advocated. Certain stresses are set up in annular linings

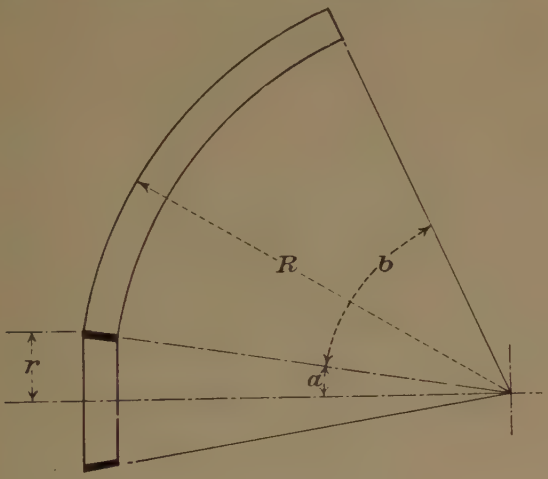


FIG. 1.

SIZING CLUTCH LINING.

of complex and unequal character, and those of an elastic nature are liable to enlarge their diameter by the effect of slipping at different velocities across the width of the face, while those of a harder and less resilient character may distort and fracture.

Obviously, therefore, floating linings should be contained on the outer diameters in a recess equal in depth to about half the thickness of the lining.

Where the clutches are exceedingly small floating linings may be satisfactory, although even then they

cannot be worn so thin as a fixed one of equal size, before fracture occurs.

Fixed Linings. Fixing linings may add slightly to the cost of production, but the increased cost is amply justified.

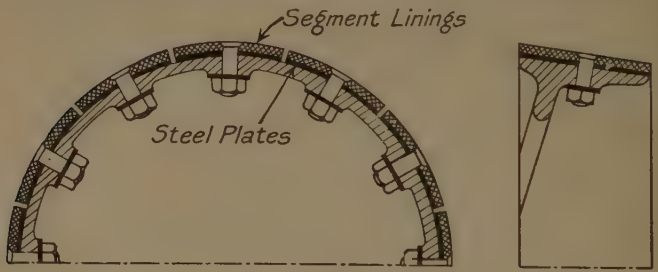


FIG. 2.
SEGMENT CLUTCH LININGS.

In cone clutches using leather linings, the linings may be renewed fairly often in the life of a clutch, consequently in the effort to make such renewal a simple matter there are certain divergencies in design.

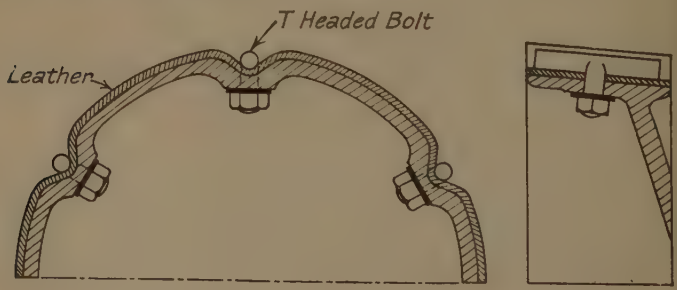


FIG. 3.
STRAINED CLUTCH LINING.

In Fig. 2 is illustrated a construction wherein the inner cone surfaces are constructed with a series of leather faced segments. The linings are built up on thin steel plates with studs attached, which are slipped into the inner

cone, which is slotted to receive them. The linings are riveted on, and as each segment is separately detachable renewal is not a difficult matter.

Another idea for fixing leather linings is shown in Fig. 3, consisting of T headed bolts fitted into grooves, which permit of the leather being well strained, thereby ensuring freedom from inequalities.

Experience of innumerable methods of fixing linings proves that there is nothing like riveting, especially for discs, providing the work is properly done.

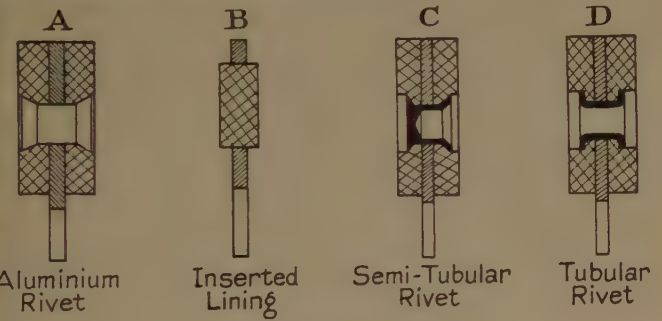


FIG. 4.
METHODS OF FIXING LININGS.

The rivets should be sunk well below the surface, and should be made of copper, or preferably aluminium.

Further, in regard to disc linings, it is advisable to rivet the linings to the driving members in preference to the driven plate. Where the driven plate is thin, it may be advantageous to rivet discs thereon, because it is advisable to insulate this as far as possible from distortion due to heat and from further reduction in thickness by wear.

Fabric linings are low conductors of heat: therefore riveting to either side of the plate, while adding to the weight of the revolving member, gives it a considerable measure of protection. In such cases tubular rivets, as in Figs. 4C and 4D, are the most convenient, but for general

practice nothing is superior to taper-headed aluminium rivets, as in Fig. 4A.

Inserted Linings. Where space is considerably limited, as in motor cycle friction clutches, the method of inserting linings has become common practice. It is, however, difficult to devise a method of retaining these inserts securely in place.

These inserts are made in various shapes, but they always tend to work loose. The circular type, as shown in Fig. 4B, rotate on their axes, against the thin edges of the surrounding metal, and then get cut or wear away more quickly than desirable.

Further, some types of insert burr over in service and the plate itself probably never comes into engagement, with consequent loss of efficiency where full load is arranged to be taken over the whole surface of driving plate.

CHAPTER III

THE CONE CLUTCH

THE cone clutch is one of the simplest forms of friction clutch that can be devised, and if correctly designed will give very satisfactory results.

Cone clutches may be grouped into four sub-classes, as follows—

1. The direct cone.
2. The inverted cone.
3. The double cone.
4. The multi-cone.

The Direct Cone. Is the most common of these types, and is illustrated in Fig. 14. It will be seen that the driving member is formed with an internal cone, into which the clutch cone is forced by the spring pressure, the spring being located behind the clutch cone.

The Inverted Cone. Is also another type in general use, and is illustrated in Fig. 16. In this type the driving member is formed in two parts, having as before an internal cone, in which the clutch cone is engaged by spring pressure, the spring being placed between the driving member and the clutch cone.

The Double Cone. Is of the type incorporating two pairs of friction surfaces, and in principle may be a combination of the two previous types, as shown in Fig. 17, or may be so designed as to necessitate the use of two clutch cones of either the direct type, or inverted type, as illustrated in Fig. 16.

The Multi-Cone. Is of the kind wherein a number of conical contact surfaces are in operation, an example of which is shown in Fig. 18.

Cone Clutch Calculations. All friction clutches depend on the contact of a driving and a driven member, the pressure necessary for a given torque being determined

2—(5298)

by the nature of the friction surfaces, that is the coefficient of friction.

In designing the clutch it is initially necessary to obtain the power to be transmitted, deducing from this the circumferential resistance to be provided at the contact surfaces.

In the following formula let—

T = Torque in lbs. at radius.
V = Velocity in feet per minute at radius.
r = Radius in inches at which T is applied.
N = Revolutions per minute
F = Axial pressure.
θ = Angle of the cone (to axis of clutch).
μ = Coefficient of friction.
W = Width of clutch face in inches.
HP = Horse-power to be transmitted.
P = Total pressure on clutch face in lbs.
$P/\square$ = Unit normal pressure on clutch face in lbs. per square inch.

Then $$HP = \frac{TV}{33000} = \frac{0{\cdot}5250\,T\,r\,N}{33000} \quad . \quad . \quad . \quad (1)$$

Where $$V = 2\pi r N \div 12 = 0{\cdot}5250\,r\,N$$

And $$T = \frac{HP \times 33000}{2\pi r} = \frac{HP \times 63000}{rN} \quad . \quad (2)$$

This represents the circumferential resistance to be overcome, and is unalterable, no matter what type of clutch.

In Fig. 5 is shown a diagram of a cone clutch, wherein the principle of the wedge is in evidence, the axial pressure being denoted by F, pressing the cones together, and P the resultant pressure acting at the contact surface.

From the force diagram we have—

$$\frac{bc}{c} = \frac{P}{F} = \text{cosec } \theta$$

Now if μ = coefficient of friction, then in order to transmit the power $P\mu$ must be equal to or greater than T.

Since $P = F \text{ cosec } \theta = \dfrac{F\mu}{\sin\theta}$ therefore

$$T = P\mu = \frac{F\mu}{\sin\theta}$$

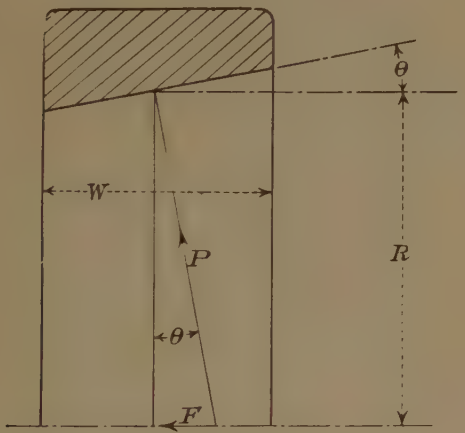


FIG. 5.
COMPOSITION OF CLUTCH FORCES.

Substituting for value of T we then obtain—

$$\frac{F\mu}{\sin\theta} = \frac{63000 \times HP}{r \times N} \quad \text{and}$$

$$F = \frac{63000 \times HP \times \sin\theta}{r \times N \times \mu} = \frac{T \sin\theta}{\mu} \quad . \quad . \quad (3)$$

There remains still to be decided the width of the contact surface. The formulae for this may be found from—

$$P = \frac{F}{\sin\theta} = \frac{T}{\mu} \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

and $W = \dfrac{P}{P/\square'' \times 2\pi \times r} \quad . \quad . \quad . \quad . \quad . \quad (5)$

From the above it will be noticed that the smaller the angle the less the force F, consequently the smaller the thrust tending to force the cones apart when declutching. However essential it may be to keep the axial pressure F at a minimum, it will be evident that this factor is determined by three other factors—

1. The frictional resistance between the contact surfaces (μ).
2. The mean radius of the clutch cone (r).
3. The angle of the cone (θ).

Distribution of Pressure. In any design of cone clutch, the unit of pressure having been arranged, it is assumed that the pressure so applied is uniformly distributed.

This is a point which requires more than passing consideration. When the clutch is new and the contact surfaces machined and fitted correctly, it is possible that the distribution of pressure is nearly uniform. However, after the clutch has been in service for a period of time, there will be a re-distribution of the pressure due to unequal wear caused by different velocities along the surfaces in contact.

It is obvious therefore that to obtain the ideal cone clutch good workmanship is essential, and that self-alignment of the contact surfaces should be incorporated in the design where possible. This will render a uniform distribution of pressure more nearly possible, and increase the durability of the clutch.

Cone Angle. Is somewhat dependent upon the materials used as contact surfaces. Cone faces faced with leather are given an angle from 8° to 13°, but the most common angles are 12° and 12½°. With metal to metal surfaces an angle of 10° may be used without risk of trouble. Using Ferodo as one of the contact surfaces, cones having an angle of up to 18° may be used without involving excessive end thrust.

It is obvious that angles too small will cause the clutch to stick, while angles too large will increase the end thrust. This will be realized on reference to Fig. 5 wherein it will be evident that the spring pressure F required to produce

a certain normal pressure decreases as the angle of the cone decreases. It will be found that an angle of $12\frac{1}{2}°$ will satisfy nearly all requirements, and it is suggested that such an angle should be standardized generally.

Provisions for Smooth Engagement. In order to secure smooth clutch engagement, devices have been originated to ensure evenness of contact between the contact surfaces.

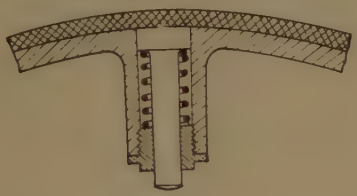


Fig. 6.
Spring Plunger under Clutch Lining.

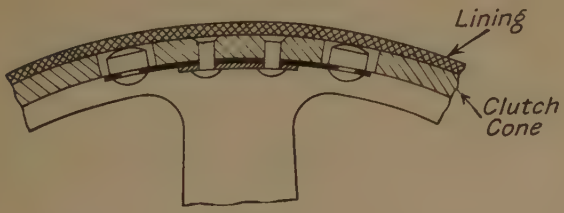


Fig. 7.
Flat Spring Plunger under Clutch Lining.

In general it may be said that the function of these devices is to raise slightly the cone facing at intervals around the periphery, so that upon engagement only a small portion of the friction surface comes into contact.

As soon as the full spring pressure is exerted the facing is depressed, and the entire surface of the cone becomes effective. One disadvantage of such devices is that they tend to increase the "spinning" effect, due to adding extra weight to the periphery of the cone.

In Fig. 6 is illustrated a device used for obtaining gradual

engagement by a spring operated plunger under the lining; while in Fig. 7 is shown an alternative method, using a flat spring instead of a helical.

Two other methods of flat cushioning springs are shown in Fig. 8 and Fig. 9, where it will be seen the spring is placed in grooves formed on the outside diameter of the internal clutch cone, or alternatively riveted to the inside of the external cone.

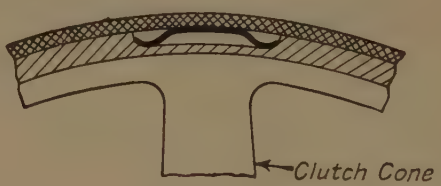


Fig. 8.

Loose Flat Spring under Clutch Lining.

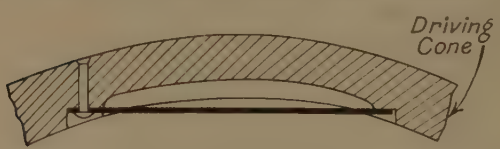


Fig. 9.

Clutch Cushioning Spring fitted to Driving Cone.

Another device for the same purpose is illustrated in Fig. 10, which consists of cutting slits either in the external or internal cones. The slits are cut lengthwise and circumferentially, as shown, and the lips are then bent inwards or outwards, as the case may be. If this method is adopted on the clutch cone it has at least the advantage of not increasing the weight.

The devices described may mitigate the engagement to some extent, but they affect the durability of the linings adversely, as those portions of the lining directly above the springing arrangements take the brunt of engagement and wear more rapidly.

The requirement is, of course, automatic cushioning

of the spring when the clutch is returned after withdrawal. To obtain this arrangements have been made to fix a pump by means of levers to the clutch centre, while the adoption of the dashpot principle, as shown in Fig. 11, presents some interesting features.

It will be seen that the dashpot is shown annular, and is fixed to the driving portion of the clutch, the annular piston and dished leather being suitably mounted thereto.

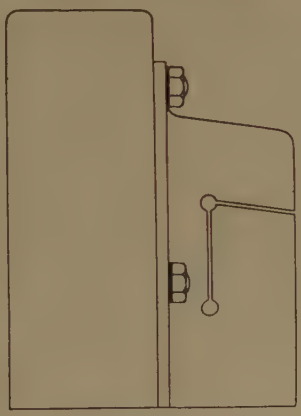


Fig. 10.
Slotted Driving Cone.

As it is necessary that the relatively small longitudinal movement of the clutch cone when withdrawn shall cause enough air to enter the dashpot for compressing purposes, it is evident that the leather should be as nearly as possible right home when the clutch is engaged. Hence recesses are provided in the bottom of the dashpot to accommodate the edges of the dished leather.

Provision is also made for the escape of the compressed air by the use of small pipes communicating with the dashpot through the dished leather, and discharging through the rim of the clutch cone to the underside of the clutch lining, thus acting as a first cushioning spring, the air escaping after performing its work.

The objections against this system may be summed up as follows—

1. The effect of centrifugal force on the leather would tend to increase the spin of the clutch cone.
2. Increased weight of the inner cone.

These disadvantages do not seem insurmountable, and the clutch might have a definite field of application, but

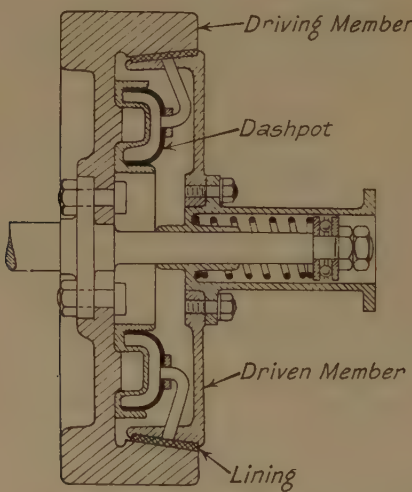


FIG. 11.
CLUTCH WITH DASHPOT ACTION.

in reference to automobiles, where it is frequently necessary to let the clutch in quickly under some circumstances, it is probable that dashpot action is not desirable.

The Clutch Cone. It is desirable to keep the diameter of the clutch cone to a minimum consistent with the desirable normal unit pressure. This applies particularly to automobile clutches where the speeds of transmission are high.

The jaw clutches of the sliding gears in the change speed mechanism of automobiles never run at similar

angular speeds just previous to being engaged, but from the instant of engagement they do run at the same velocity.

Obviously at the instant of engagement one of the engaging parts changes its speed suddenly. Also one of these parts is mechanically connected to the driving wheels of the vehicle, and cannot therefore change its speed quickly.

The other engaging part is directly connected to the clutch cone, and the resistance to a change of speed of these parts is proportional to the sum of their polar inertias, of which the inertia of the clutch cone is by far the greatest.

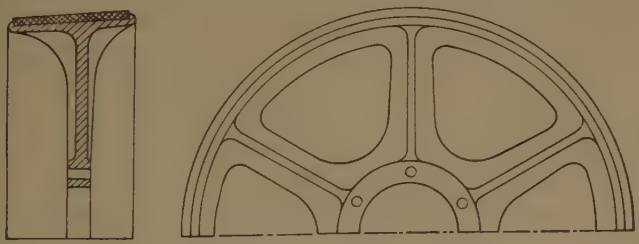


FIG. 12.
ALUMINIUM CLUTCH CONE.

The inertia of a revolving mass is proportional to its weight and to the square of its radius of gyration, which latter in a clutch cone is its outside diameter, hence this diameter must be kept as small as possible.

Again, the clutch cone diameter must be such as will ensure a reasonable unit normal pressure and the total spring pressure at a desirable amount.

For automobile work it is essential to keep down the weight of the clutch cone, to reduce the "spinning" effect when disengaged for purpose of gear changing.

For this reason the clutch cone is frequently made of aluminium, as shown in Fig. 12, its periphery being lined with a friction material.

Pressed steel cones can also be made suitable for this purpose, but care must be taken that the gauge of the

material is such as will ensure sufficient rigidity. It may therefore be desirable to press the cones with radial ribs for strength.

An example of a pressed steel cone clutch is illustrated in Fig. 13, and as will be seen it is an exceedingly light construction.

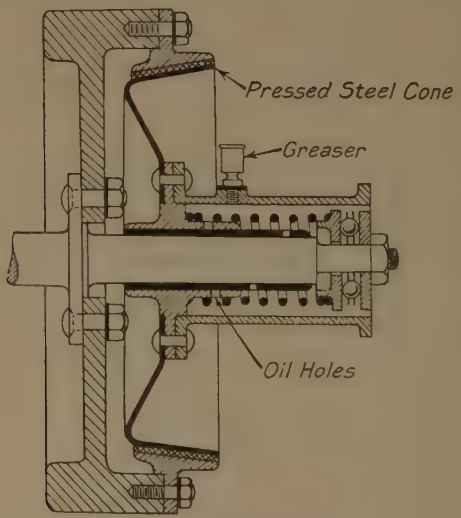


FIG. 13.

PRESSED STEEL CONE CLUTCH.

The Cone Clutch Centre. Usually the cone clutch consists of three main parts, viz., the cone, the bearing supporting the cone, and a spring housing or hollow shaft for transmitting the motion. Frequently the cone and its supporting bearing are made integral for the purpose of weight reduction, and in such instances special provision has to be made for the bearing.

The clutch bearing is only in operation when the clutch is disengaged, consequently it carries only light loads, and is usually a plain bearing dependent for its lubrication

from a greaser feeding through holes and grooves, as illustrated in Fig. 13. The clutch spring usually surrounds the bearing, the pressure being self-contained.

For the purpose of increasing its strength the web and arms of the clutch cone are inclined backward, consequently the flange connecting it to the clutch centre may be at a considerable distance from the driving cone support. Hence it is advantageous to design the bearing as far forward as possible, with due allowance for wear of the clutch, or, in the case of the inverted cone, for the necessary clutch movement.

In designing the centre of the clutch the question of assembly requires careful attention. In most designs the clutch spring housing extends to the supporting spindle. The spring must therefore be mounted and adjusted before the housing is erected.

In some instances the web of the cone is bolted to the flange, and intermediate bolts pass through the web and the flanges of both the bearing and the housing.

This method permits of assembling the cone with the bearing, placing this on the supporting spindle, erecting the clutch spring, and finally bolting up the clutch spring housing to the cone and bearing.

An alternative method is to arrange the web of the cone to come between the flange of the bearing and the flange of the spring housing, and pass all the retaining bolts through all three connected parts.

This renders it necessary to erect these parts after the spring has been mounted, which can only be done when the clutch cone is formed with arms. In some cases the clutch housing is connected to the clutch bearing by means of radial studs or bolts.

Automobile Cone Clutch (*Single Spring*). In Fig. 14 is shown a form of clutch in common use, where it will be noted that the driven cone is forced into engagement with the flywheel rim by pressure of the helical spring.

The cone is of cast aluminium, having its periphery lined with leather or fabric. The extreme simplicity of this design is apparent.

Automobile Cone Clutch (*Three Springs*). Another

example of the single cone clutch is illustrated in Fig. 15. The outstanding features are the pressed steel driven cone and the method of producing pressure by three springs. The latter has the advantage that the springs can be adjusted or replaced without disconnection of the clutch shaft and operating mechanism.

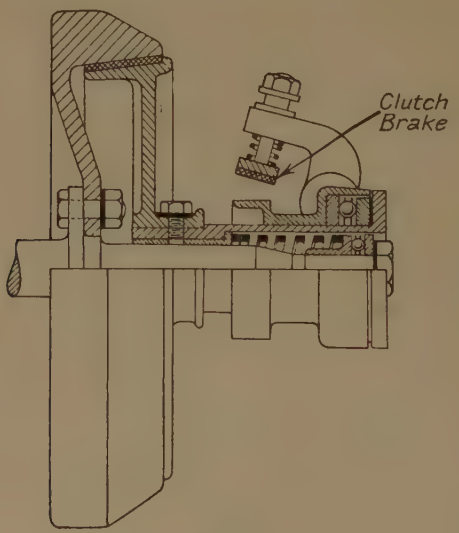


Fig. 14.
Automobile Cone Clutch.

The Inverted Cone Clutch. Apparently the only advantage which accrues from this design is that of compactness.

An example is illustrated in Fig. 16 where it will be seen that it is necessary to form the driving member in two portions, i.e. the flywheel and the external cone bolted to it.

In this construction the extended spindle of the driving shaft of the direct type is eliminated, the clutch shaft

being extended into a bearing carried by the driven member. All other details are carried out in a manner similar to those previously shown.

It will be noted in this design that the spring tension is adjustable by a screwed nut, which can be operated through the arms formed in the clutch cone.

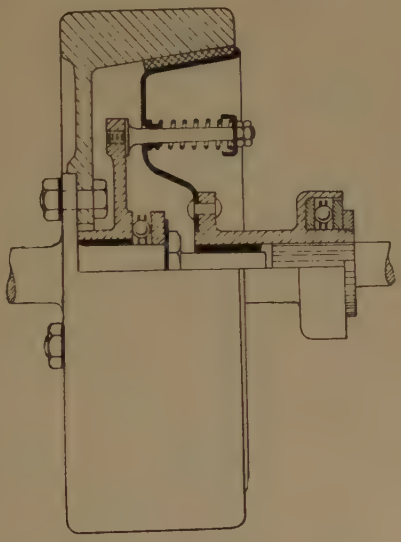


FIG. 15.
MULTIPLE SPRING CONE CLUTCH.

In this instance assembly is extremely simple. Ball bearings are used for journal and thrust loads, and the use of cumbersome couplings is obviated.

The only point against this type of cone clutch appears to be a possible restriction of cone diameter.

The Double Cone Clutch. The general principles of the double cone clutch are in every way similar to those previously outlined. The objectives aimed at in the design of a double cone clutch are reduction in diameter

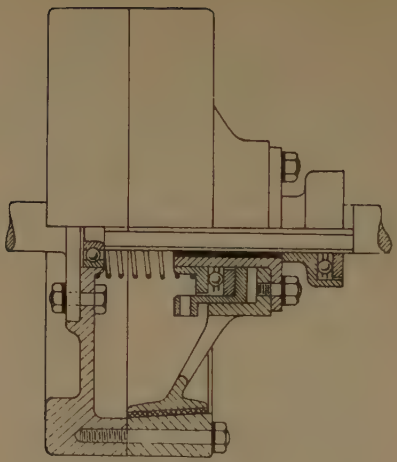


Fig. 16.
Inverted Cone Clutch.

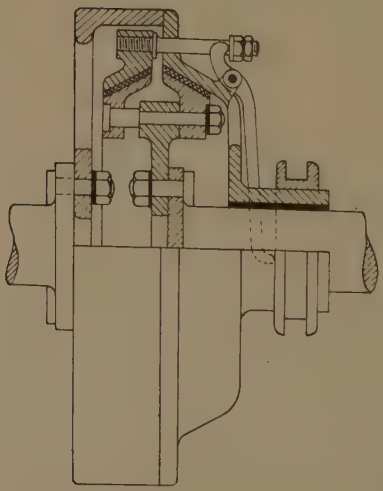


Fig. 17.
Double Cone Clutch.

of the clutch cones and elimination of end thrusts when the clutch is either in or out of action.

It is obvious that two cones transmitting the same power as a single cone would need to be considerably less in diameter, which would also result in weight and centrifugal effect being reduced.

In Fig. 17 is illustrated a clutch of this type recently

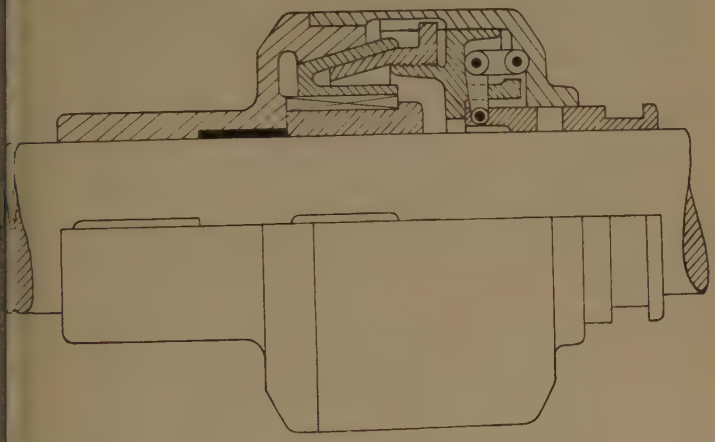


Fig. 18.

Multi-cone Clutch.

patented, where it will be seen that the driving shaft carries two external clutch cones, one of which is constructed in such a manner as to form a bearing for the clutch shaft, so that the latter will be perfectly co-axial with the driving shaft.

The clutch shaft is provided with a flange, to which is bolted an annular disc, with holes arranged for the reception of bolts.

The driving cones are connected to each other by bolts, which allow of axial float, and have mounted upon them springs normally acting to separate the cones.

Actuating bolts for drawing the two cones together are mounted in one of the driving cones, and extend through the other for engagement with radially arranged levers, bearing upon a collar slidably mounted on the boss forming the bearing for the clutch shaft.

This collar, as usual, is controlled by a clutch spring and operating gear.

It will be evident that perfect concentricity of driving and clutch shafts is ensured, and that axial float of the flanged member ensures effective engagement without imposing end thrusts in either direction.

CHAPTER IV

DISC AND PLATE CLUTCHES

In general, disc clutches consist of a series of discs arranged in such a manner that each driven disc is located between the driving discs.

Plate clutches are based on the same principle, differing from disc clutches only in detail of design.

Disc clutches are made in a variety of forms, and are in extensive use for many classes of power transmission. They may be broadly grouped into two classes as follows—

1. Plate clutches, in which one or two plates serve as the driven member.

2. Multiple disc clutches, in which more than two discs operate as driven members.

These may be further classified in regard to whether their friction surfaces are run dry, or are lubricated, which naturally depends upon the nature of the friction surfaces.

The introduction of fabric friction linings has to a great extent made the plate clutch a real engineering proposition; particularly has this been the case in connection with automobiles.

It may be noted that clutches for this purpose have to operate under particularly severe conditions. The clutch is used to connect the engine to the transmission, and thence to the road wheels through all gear reductions.

It should also be remembered that clutches should be designed for maximum loads coming upon them, and in the case of automobiles the loads are greatest when the motor transmits the maximum torque.

Plate and disc clutches are particularly in evidence in this class of work, the advances in design having been coincident.

Calculation of Disc or Plate Clutches. In Fig. 19 is shown diagrammatically a clutch disc, in which ω is the

width of a narrow ring of radius. The area of this ring would then be—

$$A = 2\pi r\omega$$

Suppose that the unit normal pressure on the surface of this disc is ρ pounds, then the normal pressure over the above area would be—

$$N = 2\pi r\omega\rho$$

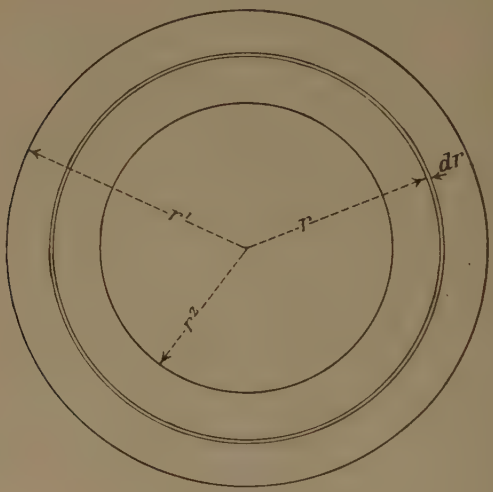


Fig. 19.

Calculation of Disc Clutches.

This causes a frictional force where μ is the coefficient of friction.

$$2\pi r\omega\rho\mu$$

and a torque of—

$$2\pi r\omega\rho\mu \times \frac{r}{12} = \frac{\pi r^2\omega\rho\mu}{6}$$

In order to obtain the twisting moment which the friction over the entire surface will produce, the above expression

must be integrated between 1 = outside radius, and 2 = inside radius. Hence we obtain—

$$\int_{r^2}^{r^1} \pi \frac{r^2 \omega \rho \mu}{6} = \frac{\pi}{18} \rho \mu \, (r1^3 - r2^3) \text{ foot-lbs.}$$

This equation will be found of service in clutches where the difference between the internal and external diameters of the contact surfaces is considerable.

In the usual type of clutch, however, the width of the contact surfaces is small, being in the form of narrow rings, the suggested reasons for which are twofold, viz.—

(1) The wear of the disc increases in proportion to the distance from the centre of rotation.

(2) The resistance to lateral motion of the discs decreases as the inner radius of the disc increases.

Dealing with the first point, obviously the speed of slippage increases with the distance from the axis ; hence if there is any considerable difference between the inside and outside radii, there will be greatly different rates of wear between the inner and outer edges, resulting in the inner edges becoming thinner than the outer, and the pressure over its surface will become unevenly distributed.

The unit pressure being greater near the inner edge than near the outer edge, the clutch will subsequently transmit less power than it would originally with the same spring pressure. Practice does not bear out this theory.

Taking the second point, it is clear that the resistance to lateral motion of the discs depends upon the pressure between the driven discs and their keys or keyway walls, which decreases as the inner radius is increased.

When the clutch is released the discs are not positively parted, but are either parted by vibration or auxiliary springs, and it is therefore desirable that the resistance to their lateral motion should be as small as possible, to eliminate " drag."

In the usual clutch, therefore, it is permissible to consider the engaging pressure and the frictional force as concentrated at the mean radius of the contact surface.

The frictional force at any contact surface is P and the

aggregate frictional force $(n-l)\,\rho\mu$ and the twisting moment or torque—

$$T = \frac{r^1 - r^2}{2}(n-l)\,\rho\mu \quad . \quad . \quad . \quad (6)$$

In any design the torque to be transmitted is a fixed quantity, but the limiting torque of the clutch is the product of four variables—

1. The mean radius of the discs.
2. The number of contact surfaces.
3. The pressure between the surfaces in contact.
4. The friction coefficient.

The above factors are independent variables, which account for the great divergence in disc clutch design.

Presuming that the materials for the contact surfaces are settled, and the coefficient of friction is then a known quantity, there still remain three variables, and the desired torque can be obtained by various means. It may be noted in this respect that an increase in the mean radius of the discs increases the inertia of the clutch, even though the number of discs is reduced to retain the same limiting torque.

Further, if the number of discs be increased, or the spring pressure augmented, the work in disengaging becomes greater.

Disengagement of the clutch is usually carried out by physical effort of the operator; consequently such work is limited in extent. Transposing the formula (6), we obtain a limiting torque—

$$T = n^s\, r^m\, \rho\eta,$$

Where n^s = number of contact surfaces,

r^m = mean radius of discs,

from which it will be evident that a measure of the work to be done in declutching will be represented by Pn^s. In practice this product is arranged to be 12,000 if declutching is to be reasonably easy.

Recapitulating the formula we have—

Area of one friction surface $= \pi(r1^2 - r2^2)$ square inches.

The frictional force between adjacent discs expressed in pounds is the same, the total frictional force at the mean radius of the discs being—

$$\frac{T \times 12}{r^m}$$

Hence the number of friction discs required is—

$$\frac{\frac{12\ T}{r^m}}{\pi(r1^2 - r\ 2^2)} = \frac{12\ T}{\pi r^m\ (r1^2 - r2^2)}$$

and the number of discs required is—

$$n = \frac{12\ T}{\pi r^m\ (r1^2 - r2^2)} + 1 \quad . \quad . \quad . \quad (7)$$

An interesting point presents itself in reference to the mean radius r^m. As shown in the formula herewith, this is represented by the arithmetical mean of the radii of the inner and outer diameter of the discs.

The correct radius at which the tractive force may be assumed to act does not coincide with the arithmetical mean, although it is near enough for most purposes.

A close approximation to the correct mean effective radius is—

$$r^m = 0{\cdot}707\sqrt{r1^2 - r2^2} \quad . \quad . \quad . \quad . \quad (8)$$

In designing a disc clutch the horse-power to be transmitted must be known, together with the speed in revolutions per minute, when the following additional formulae will cover most points appertaining—

Where HP = Horse-power to be transmitted,

R = Revolutions per minute,

N = Number of pairs of driving and driven contact surfaces,

Q = Tangential tractive force at P^m,

C = A constant $= \frac{33000}{2\pi} = 5250$,

Then $$T = \frac{HP \times C}{R} \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

$$Q = \frac{T}{r^m} \quad . \quad . \quad . \quad . \quad . \quad . \quad (10)$$

$$P = \frac{Q}{N\mu} \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Dry Plate Clutches. In order to overcome the dragging tendency of the cone clutch and the lubricated multiple disc clutch, the dry plate type was introduced and developed.

The dry plate clutch is of course a very old idea, but its practicability and success can be definitely allocated to the invention of Ferodo non-abrasive friction materials.

In the usual dry plate clutch the driving or driven plate or plates are faced with special material, of which there are a number particularly manufactured for working absolutely dry. Fabric materials of woven asbestos are the most popular, while cork and certain compositions containing rubber have been and are still used.

The question of friction materials has been previously dealt with, and the advantages of fabric materials working in a dry state adequately shown, but the main points in relation to clutches required to run dry are—that the difficulties arising from the abrasive action of dry metal to metal plates are entirely eliminated.

Dry plate clutches properly designed, utilizing these extraordinary fabric friction surfaces, will comply with every normal condition likely to be encountered. They will resist the effects of heat generated by forced slippage, are silent and flexible in action, while their working life is extraordinary in extent.

The trend in design is almost entirely in the direction of the dry plate clutch, and there is no doubt that in the immediate future this trend will be marked to the eventual exclusion of all other types. Particularly is this a fact in regard to automobiles, motor cycles, and mechanisms where high speeds obtain.

Single Plate Clutches. This type of clutch is extremely

simple in design, construction and manufacture. There is not the slightest doubt that the clutches of this type should be run dry, and although under certain circumstances it may be possible to produce a successful metal to metal lubricated example, still better results are obtainable if non-abrasive contact surfaces are utilized.

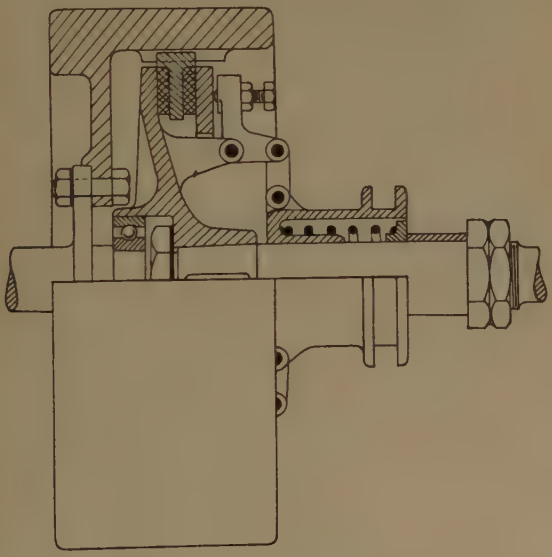


FIG. 20.
THREE-PLATE TOGGLE CLUTCH.

This type of clutch will obviate the tendency to "drag," so common in cone and multiple disc clutches.

Single (Driving) Plate Clutch. In the single plate clutch there are only two friction surfaces; hence for moderately high power transmission it is necessary to use a fairly large plate, and multiply the spring pressure by some kind of toggle mechanism.

An example of this type is shown in Fig. 20, the single

plate in this instance being the driver. One of the driven discs floats, while the other is definitely fixed to the clutch shaft. The fixed disc has lugs formed on its outer face to which bell-crank levers are fulcrumed. One arm of each bell-crank is connected by links to a sliding sleeve on the clutch shaft upon which the spring acts. The other

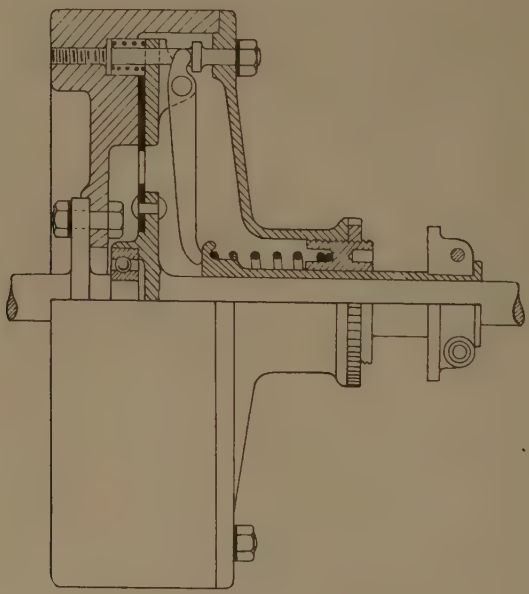


Fig. 21.
Single Plate Clutch Lever Type.

arm of each bell-crank operates against the floating disc by means of set screws. Thus movement of the bell-cranks under movement of the spring causes the driven plate to be clamped between the two driven members, the latter thereby taking up the drive.

Single (Driven) Plate Clutch. In Fig. 21 is shown a plate clutch wherein the single plate is the driven member.

In this clutch the flywheel of an automobile engine forms one of the driving members, the other driving member being formed with lugs, which through the medium of pins and levers is carried on the clutch cover fixed to the flywheel. This driving member is capable of lateral movement.

Due to the action of a series of springs operating on the

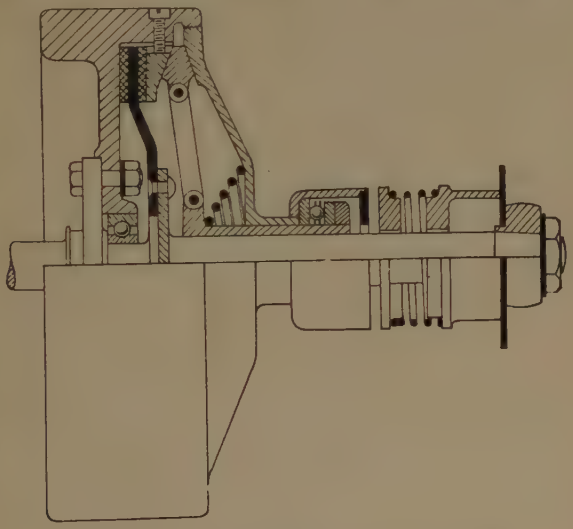


FIG. 22.
WEDGE AND TOGGLE PLATE CLUTCH (VELIE).

mean diameter of the driving members, the driven member is clamped between them.

The clutch is released by overcoming the action of the clutch springs by pressure upon the sliding collar and the levers, the lever system in this instance being utilized to reduce the spring pressure at the source of operation.

Single Plate Clutch (Wedge and Toggle). The object of this clutch, Fig. 22, is dual multiplication of the spring pressure at the contact surfaces. The flywheel face is

used as one driving member, and has the other driving member keyed to it, but free to slide. The driven plate is definitely fixed to the clutch. It will be noted that the pressure of the clutch spring, which is of the conical spiral type, is multiplied at the contact surfaces, first by the toggle mechanism, and again by the action of the wedge.

Twin-Plate Clutch. The principle of this clutch is the same as before, the only difference being the use of two

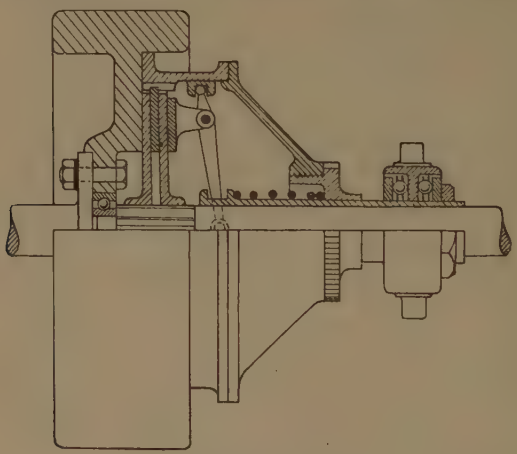


Fig. 23.
Twin Plate Clutch.

driven plates. Two plates will obviously require to be of less diameter to transmit the same power as a single plate, other things remaining similar. Reduction of the diameters will decrease the "spinning" effect, a most desirable objective.

Referring to Fig. 23, it will be seen that use is made of purchase levers for effecting and multiplying the spring pressure. The screwed ring in which the ball ends of the levers work renders adjustment for wear of the friction surfaces an easy matter.

Driven Plates. The driven member of a plate clutch, although usually of simple construction, may form the subject of considerable thought and ingenuity in design.

In plate clutches the diameter of the driven plate may be of fair size, and for this reason there is considerable

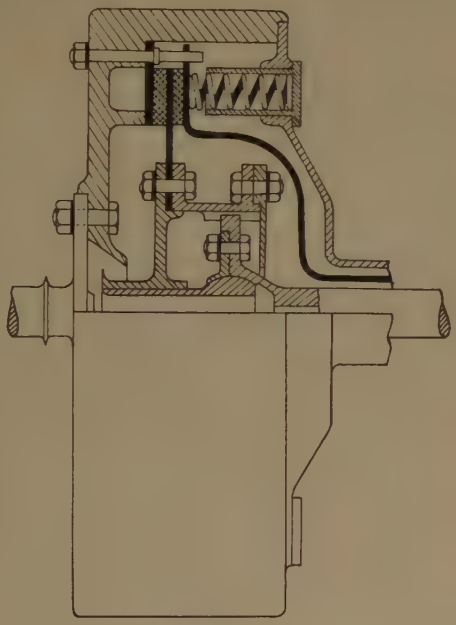


Fig. 24.

Self-aligning Plate Clutch (Albion).

divergence in design, because it is an essential factor that the faces should perfectly align themselves when coming into contact with the driving elements.

If alignment at the friction surfaces is not attained, smooth action of the clutch will not be possible while heating up, and stressing of the various members will ensue.

In Fig. 24 is illustrated a spherical aligning device, consisting of a ball joint, upon which the sleeve carrying the boss of the driven member can arrange itself to any circumstances of contact at the friction surfaces.

In the Waring-Brown Clutch shown in Fig. 25 the

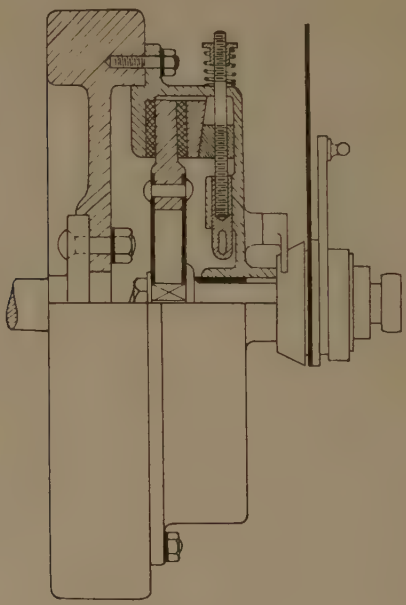


Fig. 25.

Self-aligning Plate Clutch (Waring-Brown).

driven plate is firmly clamped between two flexible three-armed plates, fixed to the centre. These flexible members are sufficiently strong to give the driven plate a rigid support, while giving enough flexibility to allow perfect alignment at the contact surfaces.

Another point in design is that concerned in keeping

the weight of the driven plate to a minimum, essential to prevent excessive "spinning" when the clutch is disengaged.

Obviously the largest diameter is the place to reduce weight, and this largely depends on the material used in the construction of the plate, which may be formed integral with its centre, as in Fig. 23, or merely made as a ring, as in Fig. 21.

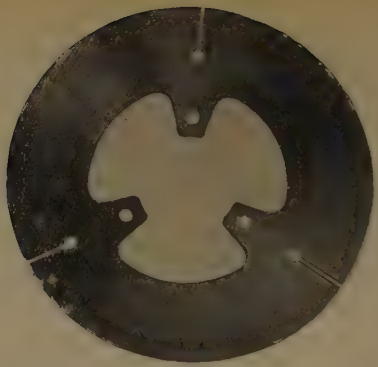
In regard to material, practically every commercial metal has been used for this purpose. Phosphor bronze under certain conditions may give satisfaction, but is subject to abrasion. Aluminium has also been used for the driven plate, on account of its heat radiating qualities, but this also quickly abrades. Cast iron and malleable cast iron have also been successfully used on medium sized clutches, the plate usually being cast integral with the centre boss.

Mild steel for this purpose is much more common, and is a practical proposition if thickness of the plate is not less than $\frac{3}{16}$ in., but if the plate is very thin it is very liable to assume a conical shape. The proper contact then not being attainable, slipping and other troubles appear.

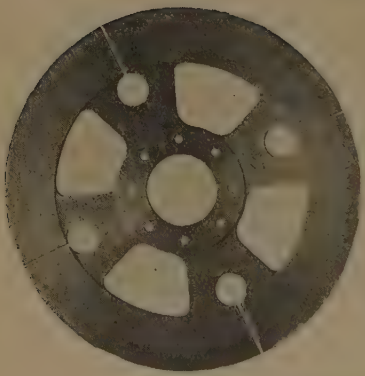
For moderate and large size plates there is no doubt that a high carbon steel is easily best for this purpose. In this material the thickness of plates up to 8 in. diameter should be not less than $\frac{1}{8}$ in., varying up to $\frac{1}{4}$ in. for diameters over 18 in.

In all cases, whatever the quality of the steel used in the construction of the driven plate, it is unquestionably an advantage to provide the plate with a number of radial slits spaced equi-distant through the outer edge across the plate, and terminating in small holes, drilled to prevent spreading. These slits may be arranged as shown in Figs. 26A, B, C, D, and prevent distortion in a most effective manner, while many minor advantages accrue from this practice.

From the points considered there appears a balance of favour in making the plate and its centre in one piece, for it must be recognized that this number is subject to sudden violent and sustained stresses, the former when

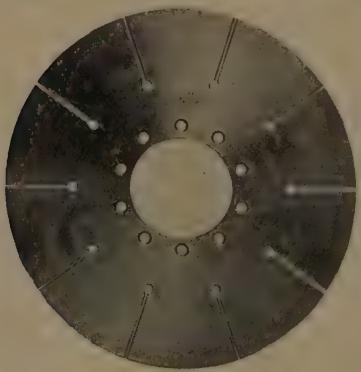


A



B

C



D

FIG. 26.

he drive is taken up from rest, and the latter all the time ower is being transmitted. Particularly is this relevant o automobiles, etc., at those times when the torque f the transmission system, arising from a free running nachine, is suddenly taken up by engaging the clutch gainst the stationary faces of the engine flywheel.

Multiple Disc Clutches. From the calculations previously iven, it is obvious that friction increases in proportion

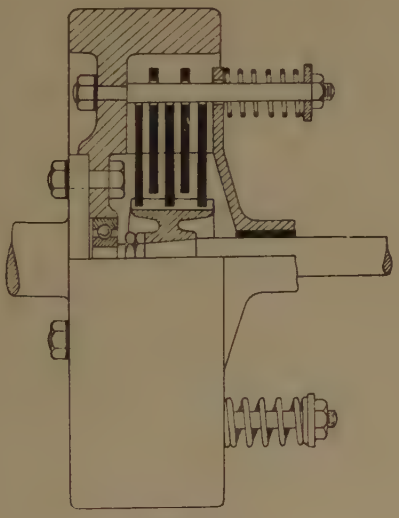


FIG. 27.
MULTIPLE DISC DRY CLUTCH.

o the number of surfaces in contact, and it is possible o increase the number and extent of these surfaces, o as to multiply the resistance due to friction to any desired amount.

As the friction surface can be increased without adding o the diameter, it will be evident that this form of clutch s particularly suitable for any purpose where the space is restricted. In practice the discs vary in number up to about sixty.

These clutches may be designed to run in lubricant or dry.

Multiple Disc (Dry) Clutch. The arguments in favour of the single plate clutch without lubrication obviously hold good if the friction surfaces be increased. In Fig. 27 is illustrated a clutch of this type. By employing a number of plates with fabric linings between them,

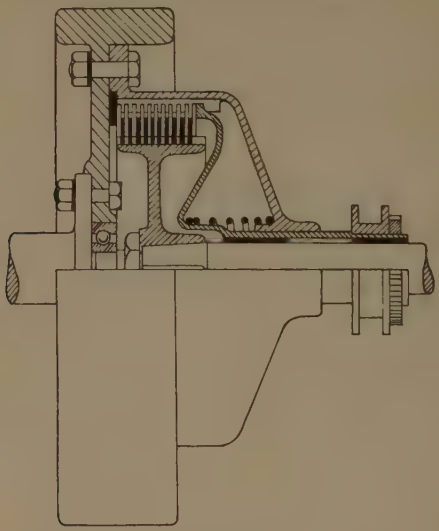


Fig. 28.
Metal-to-metal Multiple Disc Clutch.

as shown, high powers can be transmitted with comparatively low spring pressures. In this instance it will be noted that dismounting of the clutch spring is not at all an easy matter.

Multiple Disc Clutch (Metal to Metal). There are three kinds of multiple disc clutches operating in oil, the differences depending upon the manner in which the pressure of the clutch is transmitted to the back stop on the driving drum.

In Fig. 28 is shown a design in which the spring pressure is taken through the case. It will be seen that the clutch casing is provided with a number of equally spaced keys, which engage with keys formed in the driving discs. Between adjacent driving discs are located the driven discs, which are carried on keys on the driven drum. The presser is held to its work by a spring acting between it and the clutch cover. The presser is withdrawn by a sliding collar attached to its smaller end.

Construction of Discs. There are quite a number of points requiring consideration in design of discs of metal to metal clutches. Practice shows that best results are obtained by using alternate discs of phosphor bronze and steel, the former being about about 18 gauge thickness, and the latter about 20 gauge. Anything thinner than this will tend to cut and twist in the driving keys or pegs.

Owing to the heat generated in constant slippage, it is necessary to lubricate these plates to prevent abrasion and also distortion, in the latter case the oil being of quantity and quality to disperse the heat.

In letting the clutch in, this lubricant between the discs is responsible for the "dragging" tendency, and methods are necessary to allow of the oil being squeezed out if clutch action is to be lively.

To attain this object the discs are frequently made with slots extending half-way over their width, or alternatively holes are drilled through which the lubricant may escape.

To prevent "dragging" when the clutch is released, it is very necessary to introduce means for separation of the discs, and two well-known methods are shown in Figs. 29A and 29B.

At A it will be seen alternate discs are provided with ears sprung to one side. These are flattened out when the clutch is in engagement, but tend to take up position shown when the pressure is removed.

Another method of obtaining positive separation is shown at B, and consists in arranging ears on the driving discs, to which small buttons are riveted. The buttons are slightly thicker than the discs; hence the ears become

slightly sprung when the clutch is in action, and tend to force the plates apart when the clutch is put out of engagement.

Obviously the clutch spring must be made strong enough to overcome these spring devices, while still maintaining sufficient frictional force between the discs.

These separating devices cannot be described as altogether satisfactory, because the heat generated in slippage

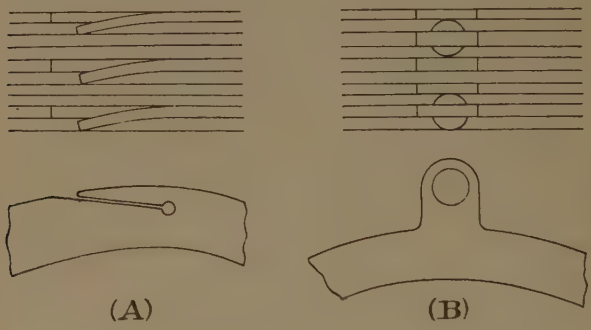


FIG. 29.
SEPARATION OF DISCS.

is liable to take the temper out of the material and permanently set the springs.

Methods of Driving Discs. One of the chief features in the design of multiple disc clutches is the method adopted for driving the discs, and alternative methods of doing this are shown in Fig. 30.

At A is shown a type wherein the plates are formed with holes for sliding on pins, which are fixed into the flywheel.

At B is illustrated the most common method, where keyways are cut in the internal diameter of the driving member, the plates being formed with lugs, which slide in these keyways. This method is more costly in manufacture, but results in a saving in weight.

At C is shown the method for driving by the bolts used

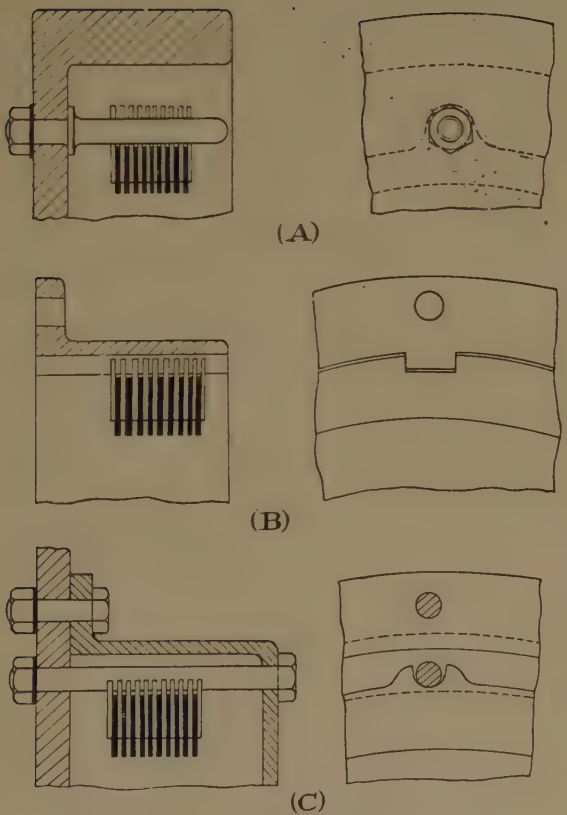


Fig. 30.

Methods of Driving Discs.

for connecting the two parts of the clutch together, the plates being formed to slide upon these bolts.

Whatever kind of driving device is adopted, there must be adequate clearance between the discs and their slides on both the driving and driven members, so that there will be no dragging when the clutch is put in or out of action.

Practice is varied as to the number of driving keys or pegs, but may be anything from three upwards.

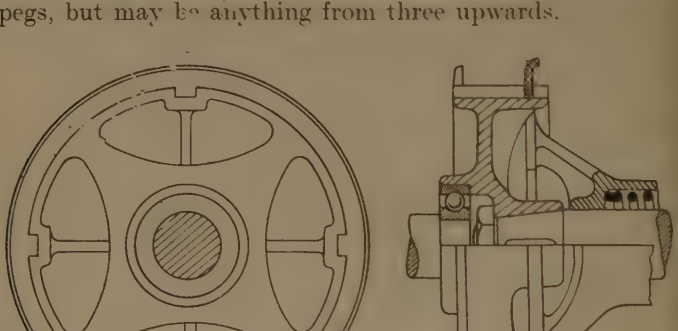


FIG. 31.
INNER DRUM FOR DISC CLUTCHES.

It is generally considered that $\frac{1}{64}$ in. is the minimum clearance between the discs which will ensure complete disengagement; consequently in designing the driving members, etc., allowance must be made for the end motion, which will be the number of plates multiplied by $\frac{1}{64}$ in.

The Inner Drum. In multiple disc clutches the inner drum is that member which takes the drive from the driven discs, and is preferably made of mild steel, formed with keyways to engage the lugs formed on the driven discs in a similar manner to those described in methods of driving discs.

The rim of the drum should be made sufficiently longer

than the combined thickness of the discs and the amount of lateral movement required for disengagement.

In order that the clutch spring will be close to its work, the spring presser is often of awkward form, and to overcome this difficulty the inner drum is made in skeleton form, as shown in Fig. 31, the rim being cut away between the keyways, the spring presser then being formed with arms entering the lateral projections of the inner drum.

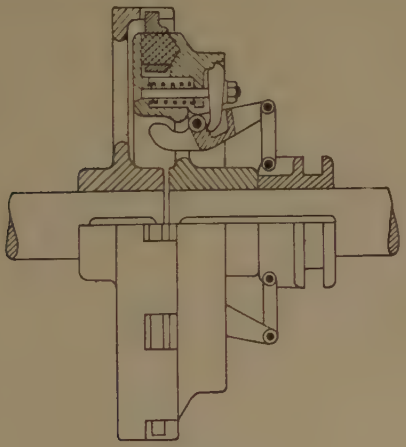


Fig. 32.
Conical Disc Clutch.

This method of design reduces the weight of the driven member considerably, and allows of a more compact construction. It is used in the flat disc and conical disc types.

Conical Disc Clutches. This type of clutch is that wherein the contact surfaces of the discs are conical. There are many clutches designed on this principle, of which a few are briefly described herewith.

An early type is illustrated in Fig. 32, and is considerably used for slow speed work. It consists of a driving member

having a slidably mounted disc fitted with hard wood blocks. There are two driven members, one with a flat face, and the other with a double cone face, which are inter-connected, and are brought into operation with driving disc through the operation of a double toggle mechanism, operated by a sliding collar. Springs are provided between the two driven members to separate them from the driving disc when the clutch is disengaged. This clutch is run dry.

The Hele-Shaw Clutch. The principle underlying this clutch is the transmission of power by frictional contact of highly lubricated wedged shaped surfaces. On reference to Fig. 33 it will be seen that the driving and driven discs have **V**-shaped annular grooves, which form the contact surfaces. The walls of these grooves are formed with angles of about 35°, and are the only parts of the discs having frictional contact, the other parts of the discs merely being used to take the drive and radiate the heat engendered during slippage.

Oil holes are drilled near the peak, through the inner walls of the grooves, to allow of easy entrance and escape of the lubricant. This design certainly allows this to be accomplished much better than the ordinary flat disc type, and consequently the action shows considerable improvement.

Corrugating the discs in this manner reduces the end thrust practically 50 per cent for a given pressure, and will transmit twice the turning effort of an equal number of flat discs of similar materials. On the other hand the discs have to be moved laterally further, to obtain necessary clearance when disengaged. Therefore the number of discs that can be used in a given space is more limited.

As in other forms of disc clutch, there is considerable "drag" effect, caused by the "skin" friction of the lubricant between the discs. Accordingly it is necessary to make provisions for separation of the discs and the introduction of a brake where high speeds are in evidence.

Conical Disc Clutch Calculations. Since the contact surfaces in this type of clutch are really frustrums of cones, the action is similar to that of an ordinary cone

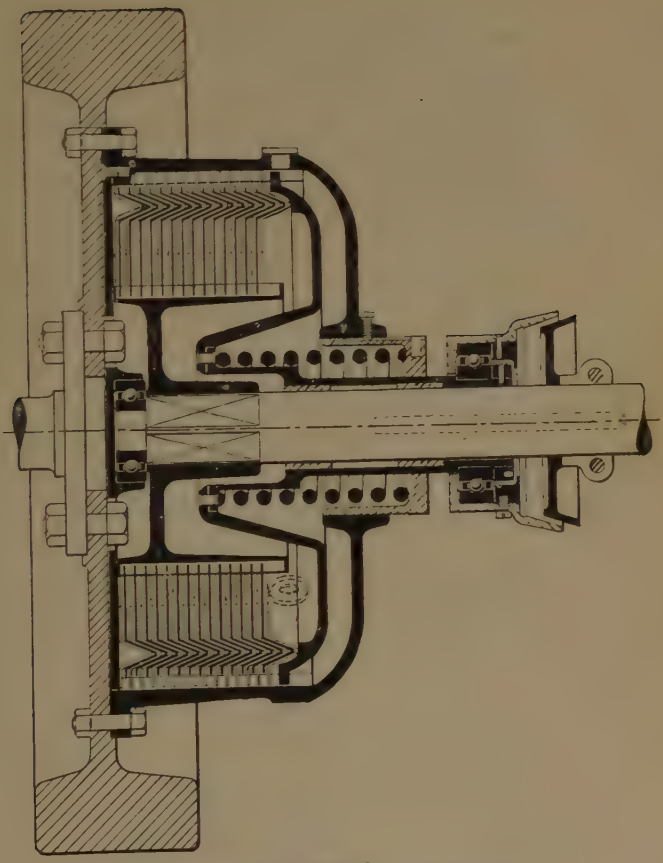


Fig. 33.

Conical Disc Clutch (Hele-Shaw).

clutch. Application of the usual principles shows that the torsional moment is as follows where—

M = Torsional moment.
n^s = Number of contact surfaces.
μ = Coefficient of friction.
P = Axial pressure.
D = Mean diameter of discs.
D^1 = Mean diameter of smaller cone.
D^2 = Mean diameter of larger cone.
θ = Half the angle of the cone.
N = Number of revolutions per minute.
HP = Horse-power transmitted.

Then $$M = \frac{n^s \mu P}{4 \sin \theta} (D^1 + D^2) \quad . \quad . \quad . \quad . \quad (12)$$

The power that can be transmitted is then found from—

$$HP = \frac{n^s \mu P N D}{126000 \sin \theta} \quad . \quad . \quad . \quad . \quad . \quad (13)$$

CHAPTER V

BAND CLUTCHES

BAND clutches are usually installed when it is necessary to transmit heavy loads accompanied by shocks, such as occur in drives for rolling mills and mine-hoists.

They are also used considerably in machine shop line shafting, but have not proved successful in automobile work, where the high speeds of rotation are against them.

In general a band clutch consists of a flexible steel band, either plain or faced with a friction material, one end of which is fixed and the other free to move in a circumferential direction. Due to the pull on the free end of the band, the latter is made to grip the driving or driven member.

There are two general types of band clutch, that is, they may be either expanding or contracting, but whichever type is considered, the principles involved are the same.

Theory of the Band Clutch. In Fig. 34 is shown a diagram of a band clutch, which will illustrate the points to be considered in reference to the pull on the band.

A small arc of contact between the band and the drum is denoted by $d\theta$, while θ represents the arc of contact between $d\theta$ and the free end of the band, where there is a pull P_1.

Owing to the friction between the band and the drum, the pull varies all round its length. The pull at one side of the differential section $d\theta$ is represented by P, and that on the other side by $P + dP$, while the normal pressure betwen the contact surfaces is denoted by N, and the frictional force resulting therefrom by fN.

When the frictional forces are in equilibrium, the forces in any direction are equal to zero ; hence taking forces on the horizontal plane—

$$(P + dP)\cos\frac{d\theta}{2} - fN - P\cos\frac{d\theta}{2} = 0 \quad . \qquad . \qquad (14)$$

But the cosine of an infinitely small angle is equal to unity; therefore—

$$dP = f N \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

Then taking forces on the vertical plane—

$$N - (P + dP) \sin \frac{d\theta}{2} \quad P \sin \frac{d\theta}{2} = 0$$

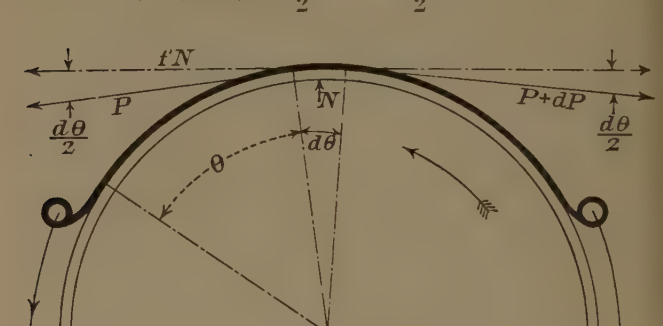


Fig. 34.

Theory of Band Clutch.

and since the sine of an infinitely small angle is equal to the arc, we have—

$$N = (P + dP)\frac{d\theta}{2} - \frac{P d\theta}{2} = 0$$

and $$N = P d\theta + dP \frac{d\theta}{2}$$

This is a differential expression, and may be written in this case.

$$N = P d\theta \quad . \quad . \quad . \quad . \quad (16)$$

and $$P = \frac{d\theta}{N} \quad . \quad . \quad . \quad . \quad (17)$$

Dividing equation (16) by equation (17) we obtain—

$$\frac{dP}{P} = f d\theta$$

Since the integral of a differential expression of this form $\frac{dx}{x}$ is log x (the Napierian logarithmic base being 2·718), then

$$\log P = f\theta + c$$

or

$$P = f\theta + \log c \quad . \quad . \quad . \quad (18)$$

As is well known the logarithm of the base is 1 ; hence—

$$\log e^{f\theta} = f\theta \times 1 = f\theta$$

Inserting this value in equation (18), we obtain—

$$\log P = \log e^{f\theta} + \log c$$

and taking antilog

$$P = Ce^{f\theta} \quad . \quad . \quad . \quad . \quad (19)$$

To find the value of constant c we make θ = zero, then P = initial pull P_1 applied to the free end of the band.

$$P_1 = c\, e^{\theta}$$

But any term with the exponent zero equals unity ; hence—

$$c = P_1$$

and inserting the value in equation (19), we obtain—

$$P = P_1 e^{f\theta}$$

This equation gives the pull on the band at any angle from the point of contact between the band and the drum nearest the point of application of the initial pull.

The total frictional force P between the band and drum is equal to the difference between the initial pull and the pull at the point of contact farthest from the point of initial pull.

$$F = F_1 e^{f\theta} - P_1 = F_1 (e^{f\theta} - 1).$$

and

$$P_1 = \frac{F}{e^{f\theta} - 1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

In using these equations the arc must be expressed in radians.

Centrifugal Force Effect. At high speeds the friction between the band and the drum is considerably affected by centrifugal force, which tends to expand the band, thereby increasing the frictional force of the expanding band, and decreasing the frictional force on a contracting band.

A section of the band 1 in. in length is denoted by ω then the weight of a portion of the band $d\theta$ is $\omega\gamma d\theta$ and the centrifugal force will be—

$$1{\cdot}226\ (\omega\gamma d\theta)\ n^2\frac{\gamma}{12} = 0{\cdot}102\omega n^2\gamma^2 d\theta$$

Where n is the speed in revolutions per second and γ the radius in inches.

This force is denoted by $F_e d\theta$ (F_e being the centrifugal force on a section of the band equal to one radian), acts in the same direction as force N in a contracting clutch hence we obtain—

$$P d\theta = N + F_e d\theta$$

Transposing and contracting this becomes—

$$(P - F_e)\, d\theta = N$$

and

$$P - F_e = \frac{N}{d\theta}$$

But since F_e is constant

$$d\,(P - F_e) = dP = f N \text{ (equation)} \qquad (21$$

Hence—

$$d\,\frac{(P - F_e)}{P - F_c} = f d\theta$$

Integrating both sides of the equation—

$$\log (P - F_e) = f\theta + c = f\theta + \log d$$

and taking antilogarithms

$$P - F_e = a e^{f\theta}$$

In order to determine this constant for this particular case—

Let $\theta = o$ then $P = P_1$

and $P_1 - F_e = a$

Hence $P = (P_1 - F_e)\, e^{f\theta} + F_e$

and $F = P - F_1 = (P_1 - F_e)e^{f\theta} + F_e - P_1$

Multiplying out $F = P_1 e^{f\theta} 1 - F_e e^{f\theta} + F_e - P_1$

$= P_1 (e^{f\theta} - 1) - F_e (e^{f\theta} - 1)$

Transposing this becomes $P_1 (e^{f\theta} - 1) = F + F_e (e - {}^{f\theta} 1)$ and dividing the coefficient by P_1

$$P_1 = \frac{F}{(e^{f\theta} - 1} + Fe. \quad . \quad . \quad (22)$$

Comparing equations 22 and 20 it will be evident that centrifugal force increases the pull on the free end of the band of a contracting clutch by an amount equal to the centrifugal force on a section of the band one radian in length.

A similar analysis is applicable to the expanding band clutch where, the centrifugal force adding to the normal pressure, the formulae then becomes—

$$P_1 = \frac{F}{(e^{f\theta} - 1)} - F_e$$

Expanding Band Clutches. As will have been seen, this type of clutch, as illustrated in Fig. 35, requires little pressure to hold in engagement, owing to the action of centrifugal force, for which reason also it is necessary to arrange for the band to be positively removed from the drum when declutching.

The band is usually made of steel, and is lined with a friction material, and supported by a skeleton drum, the engaging pressure being furnished by a tension spring, one end of which is anchored to the supporting drum.

The action of disengaging is performed by the taper key acting on to the roller, which forces the arm outwards, thus contracting the mechanism.

In Fig. 36 is shown a two band clutch, which is used for mine-hoists. The action in this instance is obtained by a toggle mechanism. The bands are lined with wood, and each band has an arc of contact of 140°.

The object of the two-band clutch is to obtain better balance, and improved contact between the bands and the drum.

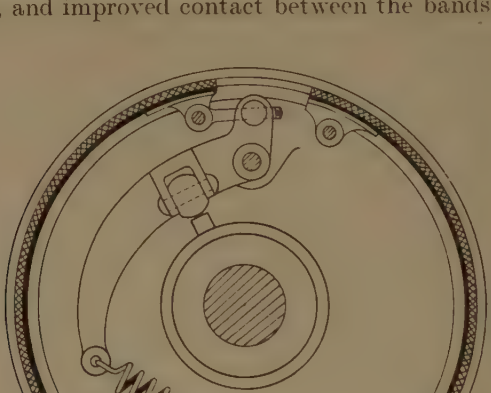


Fig. 35.

Expanding Band Clutch.

Contracting Band Clutch. A contracting band clutch consists of a drum and a band surrounding it. One of the ends of the band is fixed to a housing carried upon one of the shafts, and the other end can be angularly displaced so as to contract the band into frictional contact with the drum.

These bands may be used singly or in pairs. The latter of course allowing of better balance and contact.

A two-band type is illustrated in Fig. 37, where the two bands extend half-way round the circumference

f the drum. One end of each band is hinged to the pider arm fixed to one of the shafts, the inwardly extending rm of the lever being capable of outward movement from he clutch centre by a cone under the pressure of the lutch spring. When the levers are moved around their ins the bands are drawn tight on the clutch drum and the rive is taken up.

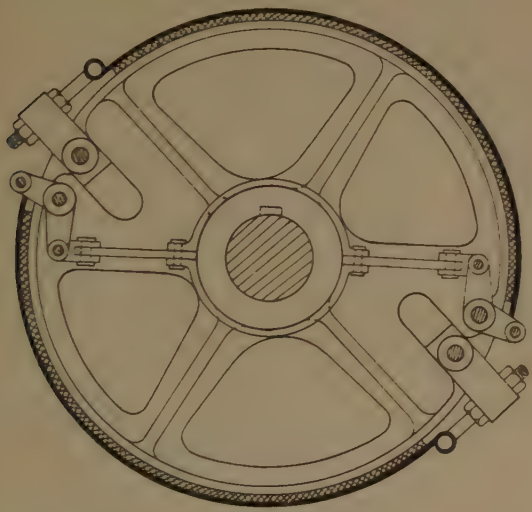


Fig. 36.
Twin-band Clutch for Mine Hoist.

Coil Clutches. This type of clutch may generally be onsidered as a form of band clutch, the band in this ase being given several turns around the driving drum.

In Fig 38 is shown one of this type adapted to automobile ransmission, in which one end of the coil is anchored to ie housing of the clutch and the other end fixed to the rm of a lever whose fulcrum support is in the end wall f the housing. The long arm of the lever is acted upon y a cone under pressure of the clutch spring. When

the cone is forced under the lever the coil is contracted upon and grips the clutch drum.

Hauling Coil Clutches. These clutches, as shown in Fig. 39 are eminently suitable for hoisting and hauling by reason of their ability to deal with high powers at very

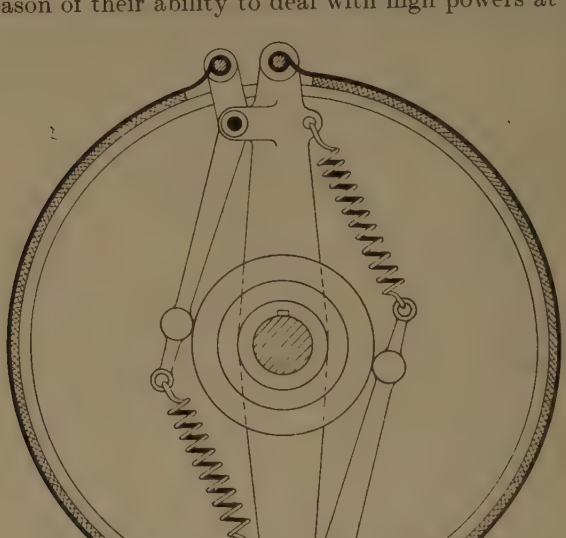


FIG. 37.
TWIN BAND CONTRACTING CLUTCH.

low speeds. It occupies very little space compared with the power transmitted, and requires very little end pressure for engagement, while being quickly and positively declutched.

The example shows a double drive clutch, which permits of power being transmitted in either direction. The general principle is as before.

Double Acting Coil Clutch. This type of coil clutch is particularly applicable to electric cranes. In Fig. 40 is illustrated the general method, which consists of a coil wound into a helic ground on its internal and external surfaces; the latter being made slightly greater than the internal diameter of the outer casing, so that the coil must

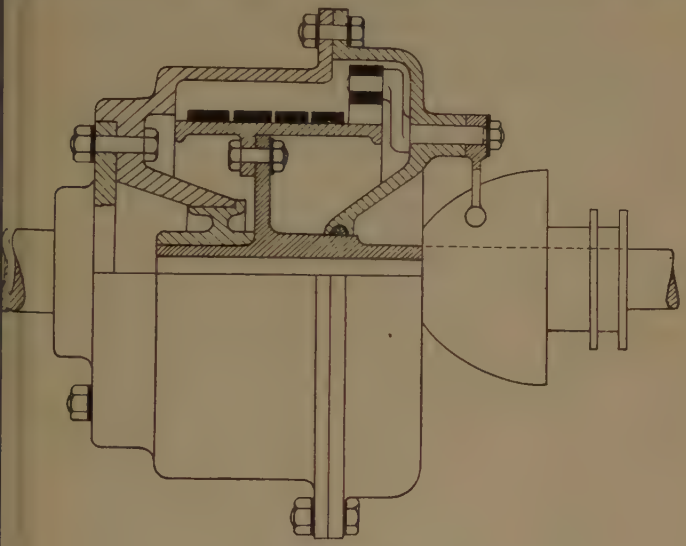


FIG. 38.

AUTOMOBILE COIL CLUTCH (MERCEDES).

be slightly compressed when assembling. One end of the coil is fixed to the flange keyed to the driving shaft, and the other end is fixed to the drum keyed to the driven shaft. The coils are usually of steel.

When correctly proportioned as to friction surface and heat radiating surface, it is exceedingly efficient.

In hoisting a load the driving shaft turns in the direction indicated by the arrow, thus expanding the coil against the outer casing, causing the driven shaft to revolve.

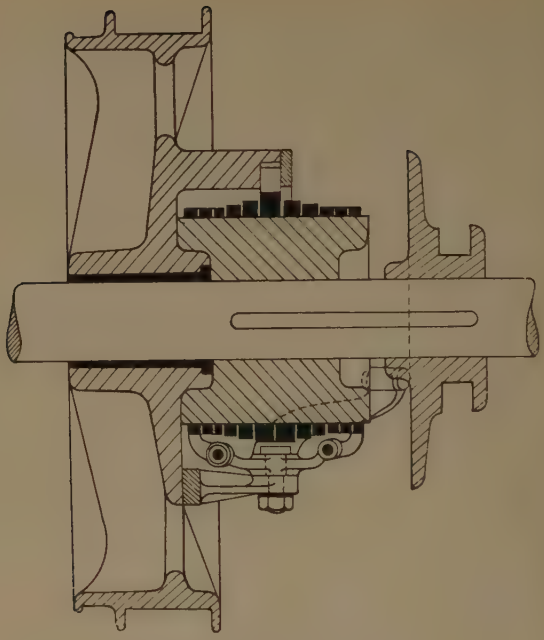


Fig. 39.
Hauling Coil Clutch.

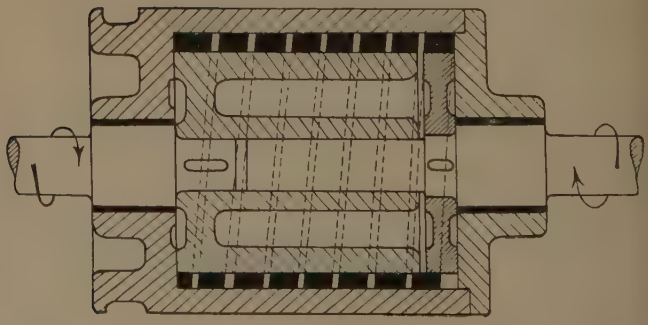


Fig. 40.
Double-acting Coil Clutch.

When holding a load suspended the driven shaft and drum turn in the direction of the arrow at the right under the action of the load ; thus again expanding the coil against the casing, which is prevented from rotating by a ratchet mechanism.

When lowering a load the driving shaft turns in the direction indicated, releasing the coils consecutively from contact. When sufficient coils have been released the driven shaft will turn under the influence of the load in the direction of the arrow ; thus again expanding the coil sufficiently to hold the load.

The ratio between the tensions of ends of the coil is denoted by K the value of which is obtained by formula.

$$\text{hyp log } K = \mu x \frac{l}{\gamma} \quad . \quad . \quad . \quad (23)$$

where μ = coefficient of friction.

l = Length of circular arc in contact.

r = Radius of drum.

CHAPTER VI

BLOCK CLUTCHES

THIS type of clutch is eminently suitable for line shaft transmission where speeds are comparatively low, for which reason they are rarely found in automobiles.

V Block Clutch. In Fig. 41 are shown the constructive features of a V-block clutch, in common use for certain classes of work. A rather objectionable feature is the exposed levers. The method of operation of this clutch is by sliding collar, which acts upon the toggle and levers, thereby multiplying the spring pressure at the contact surfaces, which is further increased by the action of wedge principle of V grooves. Obviously large powers can thereby be transmitted.

The calculations for this type of clutch are simple. From the arrangement of the spring pressure mechanism the pressure at the contact surfaces is obtained, and the frictional force is the product of the normal pressure by the friction coefficient.

In Fig. 42 is shown a diagram wherein a single block is held by force P. Then calculating in the usual manner, we find that the frictional moment that each block will transmit is—

$$M = \frac{\mu P D}{\sin B}\left(\frac{\sin\theta}{\theta + \cos\theta \sin\theta}\right) \quad . \quad . \quad . \quad (24)$$

The total moment the clutch is capable of transmitting is obtained by multiplying by the number of blocks.

Heywood and Bridge Clutch. This is an exceptionally good type of block clutch by reason of its balance, the spring pressure multiplying mechanism, large area of contact and robust construction. In Fig. 43 it will be seen that the blocks are divided transversely, but retain unity by connection to the arms, between which operate in sockets screws having right and left handed threads which are in turn connected by a lever arrangement

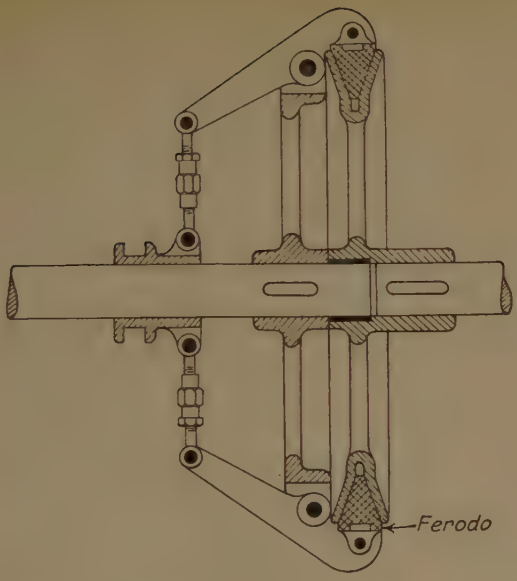


FIG. 41.
TRANSMISSION BLOCK CLUTCH.

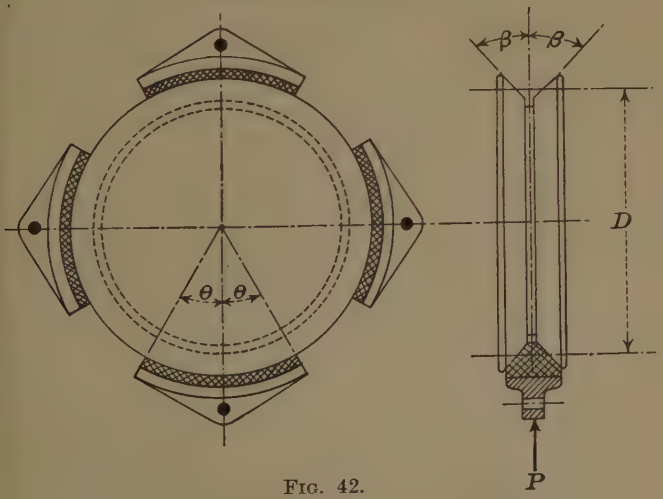


FIG. 42.
FRICTIONAL MOMENT OF V-BLOCK CLUTCH

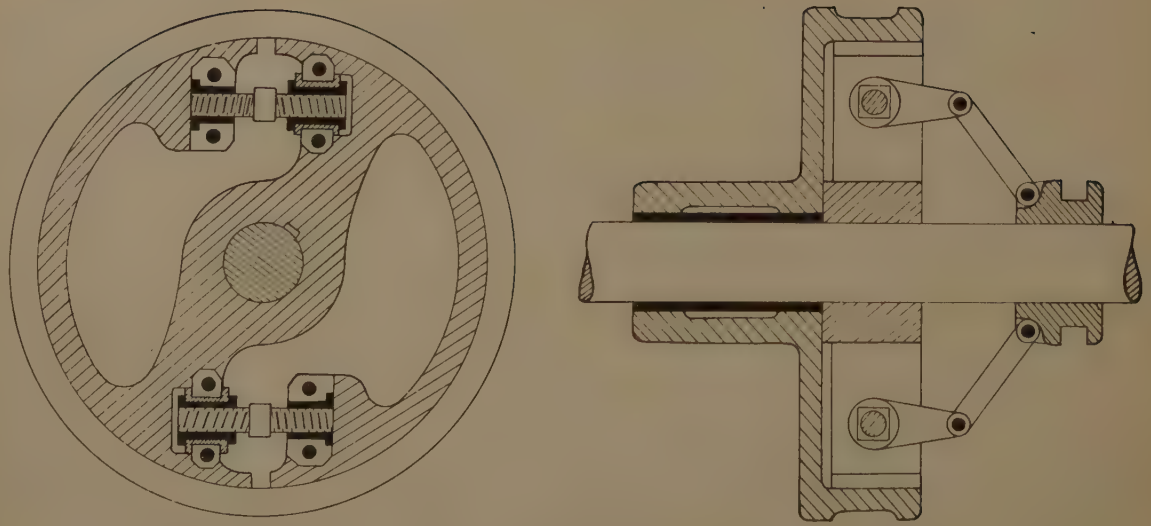


Fig. 43.

Block Clutch (Heywood and Bridge).

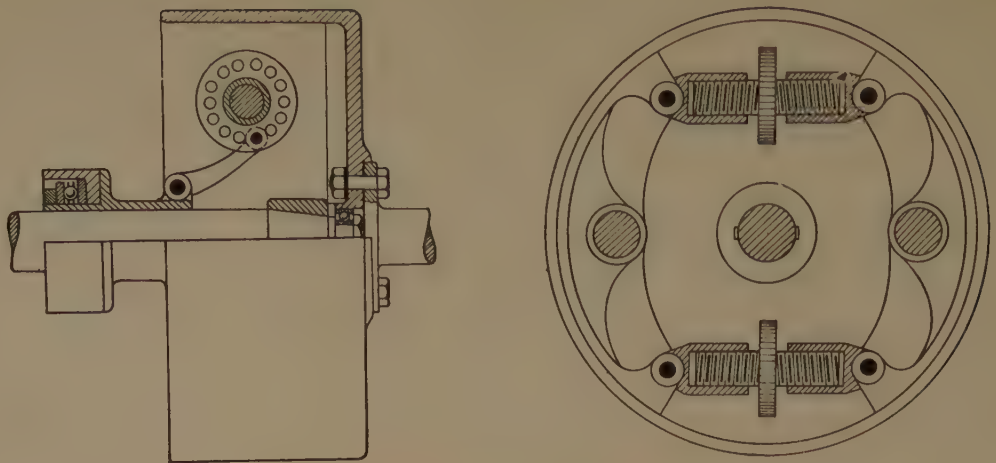


Fig. 44.

Automobile Block Clutch (Metallurgique).

to the sliding collar. Forward movement of the levers rotates the screws, and therefore thrust out the friction blocks. Obviously the axial pressure transmitted through the sleeve is multiplied by the mechanical advantage of this mechanism.

Automobile Block Clutch. This is a clutch very similar to the previous example, and is to be found on the Metallurgique car. It will be noted that the blocks consist of two shoes carried on pins mounted in the driven member, and are brought into engagement by screw mechanism exactly as previously described. This clutch is illustrated in Fig. 44.

Formulae for Screw Pressure Mechanism. This may be found as follows—

Where M = Mechanical advantage.
ρ = Pitch of screw threads.
l = Leverage centre on screws.
Q = Circumferential resistance in foot-lbs.
r = Revolutions per minute.
HP = Horse-power transmitted.
R = Radius of clutch in inches.
μ = Coefficient of friction.
F = Axial pressure in lbs.
P = Resultant pressure in lbs. at contact surfaces.

Then if screws are single-threaded $= M = \dfrac{l \times 2\pi}{\mu}$

Or ,, ,, double ,, $= M = \dfrac{l \times \pi}{\mu}$

The radial pressure required $= P = \dfrac{Q}{\mu}$

From which it follows— $P = \dfrac{l \times \pi}{\rho}$

And $Q = P\mu =$ $P\mu \times \dfrac{l\ x}{\rho}$

$$\text{Then} \quad F\mu = \frac{l \times \pi}{\rho} = \frac{63000\ HP}{R \times r}$$

$$\therefore\ F = \frac{63000\ HP}{R \times r \times \mu} \times \frac{\rho}{l \times \pi}$$

$$\text{And} \quad HP = \frac{P \times R \times r \times \mu}{63000} \quad \frac{l \times \pi}{\mu} \quad . \quad . \quad (25)$$

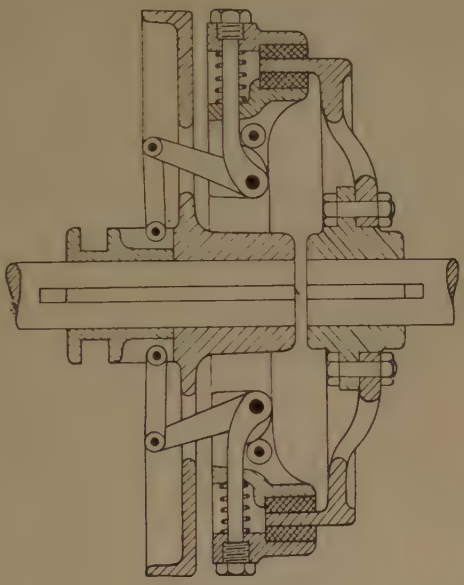


Fig. 45.
Rim Clutch (Wigglesworth).

An outstanding feature of these clutches is the small end thrusts, but the rotating mass is considerable.

Flat Rim Clutches. The majority of block clutches in common use are of the flat rim type, the Wigglesworth, shown in Fig. 45, being a fine example. It will be seen that the lever mechanism carrying the friction blocks is

located internally, thus decreasing the air resistance at high speeds. Friction blocks of suitable material operate on the external and internal diameters of the rim. The clutch is shown in gear, the operation of withdrawal being performed by a sliding collar operating cams and rollers on to the inner blocks, the resultant pressure being taken by the nuts on the bolts, which therefore pulls on the external blocks. The great mechanical advantage of this system is obvious, while in this type the frictional moment by each block is expressed in formula—

$$M = \mu P D \left(\frac{\sin \theta}{\theta + \cos \theta \sin \theta} \right) \quad . \quad . \quad . \quad (26)$$

The total amount being obtained as previously described in reference to V block clutches.

CHAPTER VII

CENTRIFUGAL CLUTCHES

Friction clutches are generally employed for connecting or disconnecting two running shafts. Centrifugal clutches, however, are designed for quite a different purpose, and are used in cases where difficulty is experienced in starting and accelerating a motor driving a load requiring a large starting torque. A particular application may be cited in reference to their use in connection with alternating current motors of the squirrel cage type.

Although this class of motor is the simplest, cheapest, and by far the most robust type of motor it is possible to produce, its application is limited by its poor starting torque. When such a motor, however, is fitted with a centrifugal clutch, it becomes as good as those motors of more complicated type, and the drive is more reliable.

The Broadbent Centrifugal Clutches. The mere simplicity of this clutch places it in the forefront. There are few parts, it is balanced, and the working parts are few in number and durable in character.

As will be seen in reference to Fig. 46, the driving member is keyed to the motor shaft, and carries with it a number of loose shoes carried in suitable pockets. Each of these shoes is lined with Ferodo on its outer face, which, owing to the high coefficient of friction, allows the dimensions of the mechanism to be kept to reasonable proportions.

As the motor when started increases its speed, the shoes are thrown radially outwards by an increasing centrifugal force, and engage with the interior rim of the pulley connected to the machinery to be driven, thus gradually accelerating this to full speed. The drive then becomes positive, as the weights of the shoes are so regulated that when the motor is running at its rated speed the power transmitted by the shoes equals the power of the motor and clip ceases.

Spring Controlled Centrifugal Clutch. For single phase motors it is absolutely essential to install spring control, owing to the low starting torque of this type of motor. In Fig. 46A is illustrated this device, which consists in instituting springs, which prevent the shoes from engaging until the motor has accelerated to any pre-determined speed up to 80 per cent of its normal rate.

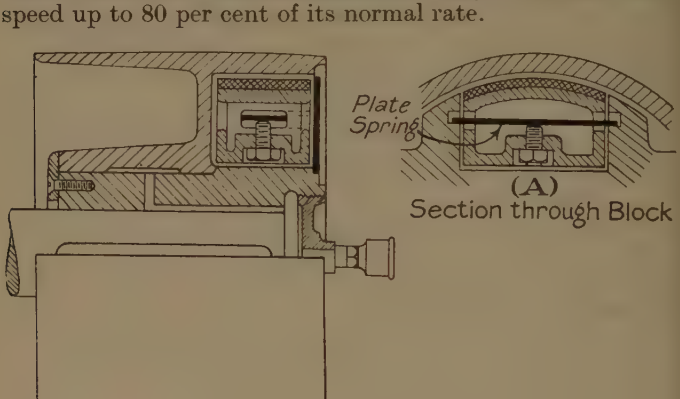


FIG. 46.
CENTRIFUGAL CLUTCH (BROADBENT).

In the "Broadbent" clutches the springs are of the flat type, and operate against the shoes through the medium of a slot in the shoe and a fixed point in the driving member, determined by adjustable set screws. The adjustment is extremely simple, and will allow for tension, time in engagement, or wear on the shoe.

Hand Controlled Centrifugal Clutches. These clutches operate in the usual manner of a centrifugal, but allow of disengagement by hand, and are advocated for use in connecting two shafts whose speeds are beyond 300 revolutions per minute.

It will be seen on reference to Fig. 47 that when the clutch is in engagement the loose shoes are free to operate

on the interior rim of the driven member, and power developed by these shoes gradually and automatically picks up the load, and accelerates the driven element to full speed. The hand control allows these shoes to be instantly withdrawn from contact with the driven member.

General Application of Centrifugal Clutches. In general these clutches may be said to be absolutely essential for

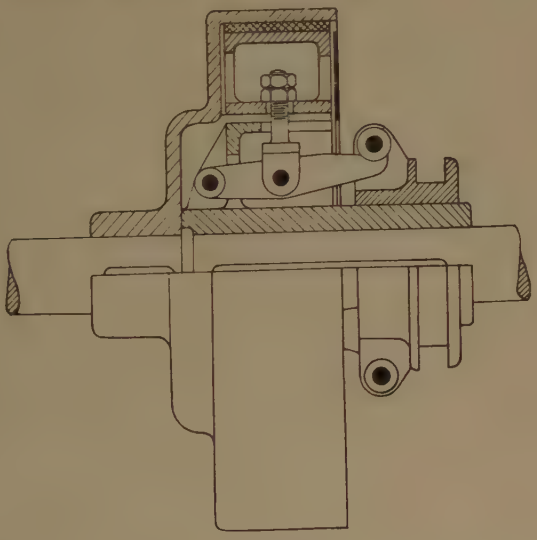


FIG. 47.

HAND-CONTROLLED CENTRIFUGAL CLUTCH (BROADBENT).

use in centrifugal machinery, such as hydro-extractors, owing to difficulties of starting and acceleration.

Again in textile machinery, machines can be started up very gradually, and breaking down of the yarn or an uneven firmness of the texture of the material can be avoided.

Apart from their adaptation to electric motors, centrifugal motors are particularly well suited to machine tool drives,

stone and ore crushers, pulverisers, haulage gears, rolling mills, and are frequently found of service in conjunction with internal combustion engines where sudden overloads are to be avoided.

Heywood & Bridge Centrifugal Clutch. This clutch is a development of the well-known patent friction clutch made by the above firm, but in this case the frictional grip

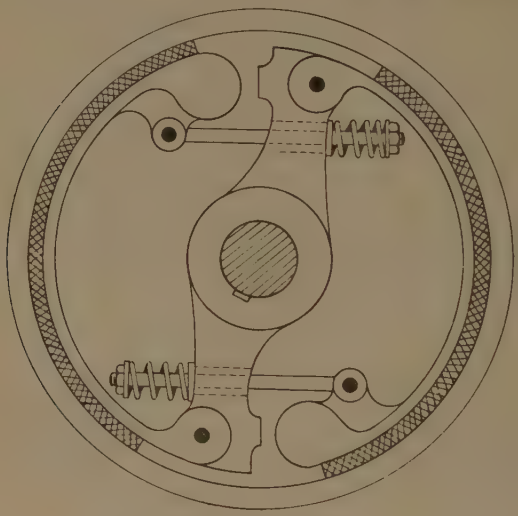


Fig. 48.
Centrifugal Clutch (Heywood and Bridge).

of the acting surfaces is produced by centrifugal force. In Fig. 48 it will be seen that the shoes are pivoted at one end of the driving member, and the action of centrifugal force in throwing out the shoes against the driven member is resisted by springs acting between the driving member and the shoe.

The method adopted obviously allows extreme ease in adjustment for picking up the load within the necessary

limits. The shoes are lined with fabric, and there are practically no wearing parts other than these.

Wigglesworth Centrifugal Clutch. This is another simple type of clutch of the centrifugal order, and is largely used in textile factories. In this clutch it will be seen that when the driving shaft has attained a certain speed, the

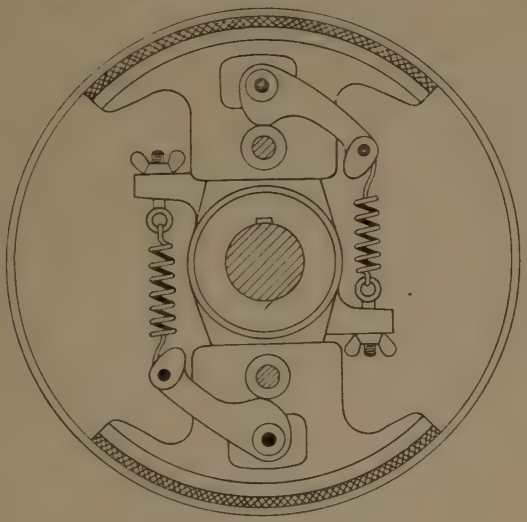


FIG. 49.

AUTOMATIC TRIP CENTRIFUGAL CLUTCH (WIGGLESWORTH).

levers fly outwards, overcoming the tension of the springs. Movement of the levers allows the slippers to come into contact with the drum of the driven member, thus automatically to pick up the load. The springs are adjustable for tension, consequently the clutch can be arranged to fully engage at any pre-determined speed. This clutch is illustrated in Fig. 49.

Slipping Clutch Couplings. In general machinery installations it is frequently necessary to install a device, which while transmitting the rated horse-power of the

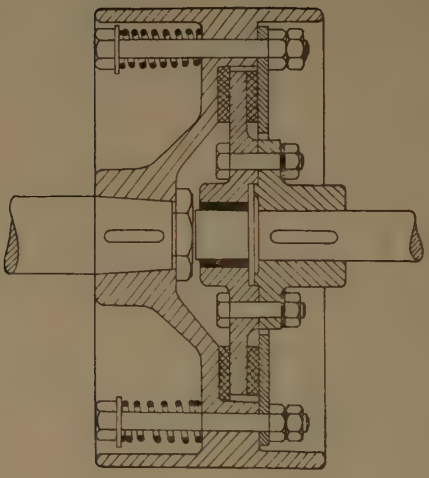


FIG. 50.
SLIP COUPLING (BROADBENT).

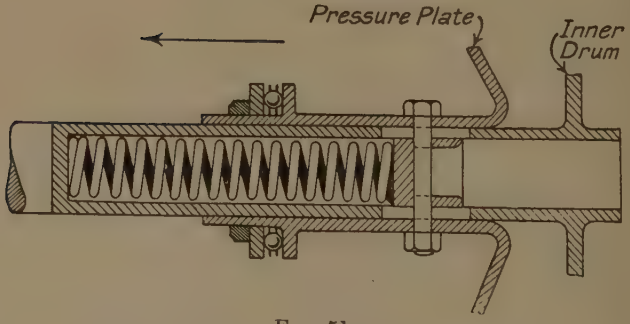


FIG. 51.
CLUTCH SPRING, INSIDE SHAFT.

prime mover, will automatically slip at any pre-determined overload, thus acting as a safety mechanism.

There are a variety of engineering applications incorporating the use of electric motors, where it is absolutely necessary to prevent damage by overload or shock.

In Fig. 50 is illustrated the " Broadbent " Slipping Clutch Coupling. A most effective device, which differs in principle from the centrifugal clutch, in that it does not ensure easy starting up of the motor, but starts up against full load torque like a solid coupling, and automatically slips under overload.

It will be seen that the device consists of a pulley fixed to the motor spindle, and a flange slidably mounted on the same spindle, together with a driving plate mounted to the pulley. The friction surfaces are " Ferodo " discs, and the pressure is applied through the medium of the springs and bolts, which clamp the driven flange between the pulley and driving plate. The torque transmitted by this device is directly dependent upon the spring pressure, and may be varied to any suitable degree within the range of the springs by adjustment of the nuts on the coupling bolts.

In Fig. 51 is shown an exactly similar coupling to above, but it has combined with it a centrifugal clutch speed governing brake, for preventing the motor from attaining a speed beyond the safe limit.

Thus the motor is protected in two distinct ways, i.e. by the clutch slipping in the event of an overload, and by braking the motor in the event of it exceeding a predetermined speed.

Wigglesworth Slipping Clutch. This clutch has been designed for use with machines liable to breakage through sudden shock or overload. Sufficient friction between the discs is ensured by the springs to take the drive under ordinary working conditions ; should the machine become suddenly overloaded the clutch will slip, and thereby the damage which would otherwise occur is obviated.

The springs are adjustable, so that the clutch may be arranged to slip at a pre-determined limit.

The simplicity of this design is apparent in Fig. 63.

CHAPTER VIII

GENERAL CONSTRUCTIONAL DETAILS OF CLUTCHES

THERE are a certain number of details which are practically common to any design of friction clutch, including such items as springs, shafts, toggle mechanisms, operating collars, clutch brakes, and bearings.

These have frequently a considerable effect on the complete design, and are therefore worthy of careful consideration.

Clutch Springs. The springs which hold the clutch in engagement are usually helical, or coiled springs manufactured from specially tempered steel wire of convenient section.

Practically every known form of spring has been incorporated in the design of friction clutches, but while the particular design involved may decide the type of spring to be used, the helical spring of circular section is eminently satisfactory for most designs.

There are three factors to be considered in relation to springs, viz.—

1. Compactness.
2. Distribution of pressure.
3. Accessibility.

These factors, of course, may not represent the relative order of merit, but are essential in practically every instance.

Spring Around Shaft. In general, it may be stated that the usual helical spring surrounding the shaft is standard design, which system allows of compactness in design, either on clutches of the cone or disc types, as shown in Figs. 13, 16, 21, 28.

In regard to distribution of pressure, it is obvious that the pressure is exerted some distance from the actual

contact surfaces, and may therefore not be so uniformly distributed as would be possible. At the same time, this is not the decisive factor.

The worst feature appears to be the fact that removal of this spring means uncoupling the shaft, which is frequently a matter of some difficulty.

Springs Inside Shaft. Some designers prefer to place the clutch spring inside the shaft. In Fig. 51 is illustrated an instance where the spring operates through a bush and a

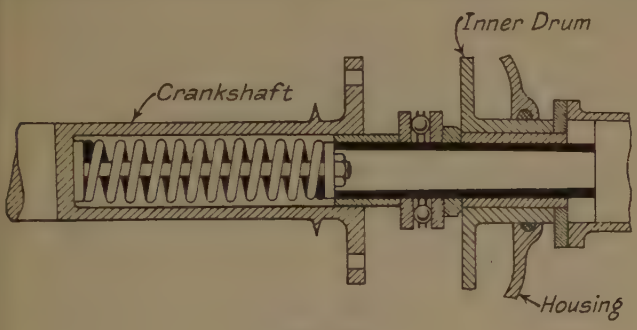


Fig. 52.

Clutch Spring, Inside Shaft.

bolt extending through a slot in the shaft on to the presser plate, the length of the hole on the shaft limiting the spring movement in the opposite direction.

An alternative design is shown in Fig. 52, which has been considerably used in automobile practice. The spring in this case is located in the rear bearing of the crank-shaft, and presses against a steel washer, and through the medium of a sleeve on the clutch shaft, a ball thrust bearing and a collar on to the inside drum. In this particular clutch the inner drum is moved in an "axial" direction to disengage.

These two designs certainly attain compact arrangement, but it entails the great disadvantages of localized spring pressure, and utter inability to adjust or renew the spring without considerable trouble.

Further, the springs are absolutely encased, and as they are subject to heating caused by clutch slippage there appears little chance of dissipation of this heat, with the probable effect of loss of spring temper.

Multiple Springs. A very common arrangement of clutch springs is that wherein a number of springs are used placed round the mean diameter of the friction surfaces in the case of disc clutches, and as near to the friction surface as convenient in cone clutches.

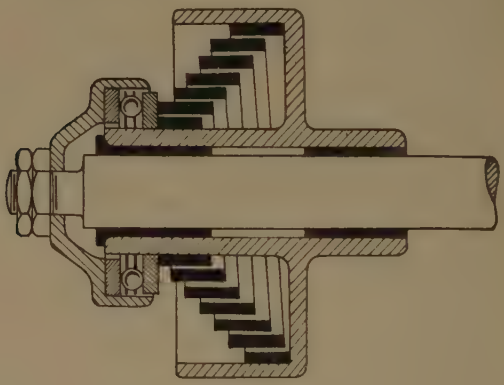


FIG. 53.

VOLUTE CLUTCH SPRING.

A specimen of this idea is shown in the cone clutch in Fig. 15, wherein three springs are used, operating on pins fastened to a thrust plate, and extending through the clutch cone. These springs are compact in design, are small, light, and cheap to produce.

Further, renewal and adjustment are simple matters, and no other parts of the clutch are necessarily disconnected. Also, the spring pressure is operating where it is actually required. Arranging the springs in this manner adds somewhat to the revolving mass when declutching, but this is not a serious item.

The chief argument against the system is that equal pring pressure on all springs may not be obtained, but in ctual practice the point is negligible.

The application of similar system to disc clutches is hown in Figs. 24, 27, 32.

Special Springs. Occasionally, in order to save space in longitudinal direction, volute springs of oblong section re used, as shown in Fig. 53, and obviously it is possible o arrange an extraordinary strong spring in a very little pace on these lines.

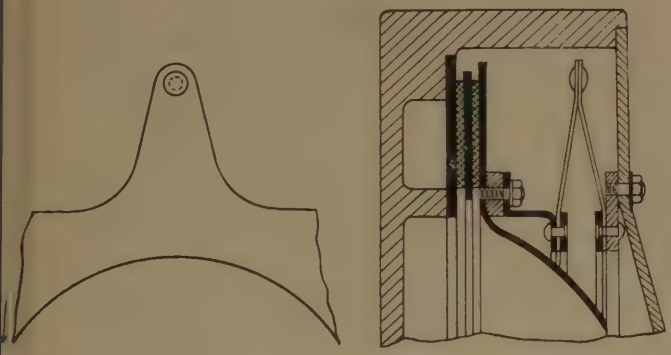


Fig. 54.

Star Plate Spring for Plate Clutch.

Flat plate springs have also been used for obtaining he necessary pressure at the friction surfaces.

A rather unusual design of plate spring is that shown n Fig. 54, wherein the spring consists of two star-shaped plates connected at their points, attached to the presser plate and the clutch cover.

The advantages of such design lie in the self-aligning qualities of the presser plate, which is entirely carried by the spring plates, and not on a bearing, which might ntroduce friction.

Clutch Spring Formulae. The calculations necessary

for the design of springs may be carried out by using formulae and tables given below where—

D = Diameter of side of section.

C = constant { for round wire = 22. / for square wire = 30.

K = Coefficient ($\frac{1}{16}'' = 17750$ and $\frac{1}{2}'' = 10000$).

R = Rate = load to compress or extend spring 1" in lbs.

N = Number of coils.

W = Maximum safe loads in lbs.

d = Mean diameter of coil in inches.

r = Mean radius of coil in inches.

Then $$R = \frac{CD^4 \text{ in } \frac{1}{16}\text{ths}}{N \times d^3} \quad . \quad . \quad . \quad . \quad . \quad (27$$

$$N = \frac{CD^4 \text{ in } \frac{1}{16}\text{ths}}{R \times d^3} \quad . \quad . \quad . \quad . \quad . \quad (28$$

$$W = \frac{KD^3}{r} \quad . \quad . \quad . \quad . \quad . \quad . \quad (29$$

$$KD^3 = W \times r \quad . \quad . \quad . \quad . \quad . \quad . \quad (30$$

To be used in conjunction with tables.

The theoretical pressure required to be exerted by the spring or springs is of course equal to the axial pressure given by previous formulae.

In actual practice it is usual to allow 25 per cent increase over the desired figure when using the allowable coefficients of friction shown in table.

After considerable service springs tend to lose some of their temper, which is the reason for making stronger than initially necessary.

Clutch Shafts. The clutch shaft may form the subject of very careful design, because of the arduous work it has to perform. The stress allowed in such shaft is very low, and a high safety factor is necessary to allow for the changes in the friction coefficient of the contact surfaces and the adjustment of spring pressure.

Due to these causes the torque transmitting capacity of the clutch may be greatly increased, and much greater turning movements than those of which the prime mover is capable of producing continuously may come upon the clutch shaft by engaging the clutch when the driving member is revolving at full speed.

Torsional Strength of Clutch Shafts. The torsional strength may be calculated from the following formulae, which has been found quite satisfactory in practice—

TABLE II

CONSTANTS FOR SPRINGS

D = Dia. or Diag. of Wire.		D^3	D^4	K	KD^3	CD^4	
Fraction or I.W. Gauge.	Dec. Eqv.	Inches.	in $\frac{1}{10}$ ths.			C = 22	C = 30
000	·400	·06400	1677·7	11600	742·40	36909	50331
$\frac{3}{8}''$	·375	·05273	1296·0	12000	623·75	28512	38880
000	·372	·05148	1255·0	12050	620·32	27610	37650
00	·348	·04214	961·17	12430	523·80	21146	28835
0	·324	·03401	722·17	12820	435·98	15888	21665
$\frac{5}{16}''$	·312	·03037	620·98	13000	394·78	13622	18629
1	·300	·02700	530·84	13200	356·40	11678	15925
2	·276	·02102	380·20	13580	285·45	8364·4	11406
3	·252	·01600	264·29	13950	223·18	5814·4	7928·7
$\frac{1}{4}''$	·250	·01562	256·00	14000	218·67	5632·0	7680·0
4	·232	·01249	189·86	14290	178·47	4176·9	5695·8
$\frac{1}{32}''$	·219	·01050	150·74	14500	152·24	3316·3	4522·2
5	·212	·00953	132·38	14600	139·13	2912·4	3971·4
6	·192	·00708	89·059	14920	105·63	1959·3	2671·7
$\frac{3}{16}''$	·187	·00654	82·005	15000	98·093	1804·1	2460·1
7	·176	·00545	62·881	15180	82·725	1383·9	1886·4
8	·160	·00410	42·949	15450	63·345	944·80	1288·4
$\frac{5}{32}''$	·156	·00380	38·881	15500	58·892	853·80	1164·3
9	·144	·00299	28·179	15650	46·792	619·94	845·37
10	·128	·00210	17·592	15950	33·494	387·02	527·76
$\frac{1}{8}''$	·125	·00195	16·000	16000	31·200	352·00	480·00
11	·116	·00156	11·865	16150	25·194	261·05	355·99
12	·104	·00112	7·6666	16340	18·301	168·67	229·99
$\frac{3}{32}''$	·094	·000829	5·0517	16500	12·970	111·14	151·55
13	·092	·000779	4·6951	16540	12·307	103·29	140·85
14	·080	·000512	2·6844	16700	8·5500	59·057	80·532
15	·072	·000373	1·7611	16830	6·2775	38·744	52·893
16	·064	·000262	1·0995	16950	4·4405	24·189	32·985
$\frac{1}{16}''$	·062	·000244	1·0000	17000	4·1478	22·100	30·000
17	·056	·000176	·64452	17090	3·0077	14·179	19·335
18	·048	·000111	·34789	17210	1·9103	7·6536	10·436
$\frac{3}{64}''$	·047	·000103	·31707	17250	1·7767	6·9755	9·5121
19	·040	·000064	·16777	17340	1·0598	3·6909	5·0331
20	·036	·0000467	·11007	17400	·81260	2·4215	3·3021
$\frac{1}{32}''$	·031	·0000305	·06299	17500	·53200	1·3858	1·8897
$\frac{1}{64}''$	·015	·0000038	·003881	17750	·067450	0·8538	0·1164

TABLE III
CONSTANTS FOR SPRINGS

d	d^3	d	d^3	d	d^3	d	d^3
$\frac{3}{32}$″	·000824	$\frac{7}{16}$″	·42818	$1\frac{7}{16}$″	2·970	$2\frac{3}{8}$″	20·887
$\frac{1}{8}$″	·001953	$\frac{15}{32}$″	·5364	$1\frac{1}{2}$″	3·375	$2\frac{1}{2}$″	23·754
$\frac{5}{32}$″	·00659	$\frac{1}{2}$″	·6699	$1\frac{9}{16}$″	4·291	$2\frac{5}{8}$″	27·000
$\frac{3}{16}$″	·01563	$\frac{17}{32}$″	·8240	$1\frac{5}{8}$″	5·359	$2\frac{3}{4}$″	30·517
$\frac{7}{32}$″	·03052	$\frac{9}{16}$″	1·000	$1\frac{11}{16}$″	6·591	$2\frac{7}{8}$″	34·328
$\frac{1}{4}$″	·05270	$1\frac{1}{16}$″	1·199	$1\frac{3}{4}$″	8·000	3″	38·019
$\frac{9}{32}$″	·08374	$1\frac{1}{8}$″	1·424	$1\frac{13}{16}$″	9·595	$3\frac{1}{4}$″	42·875
$\frac{5}{16}$″	·1250	$1\frac{3}{16}$″	1·675	$1\frac{7}{8}$″	11·390	$3\frac{1}{2}$″	42·875
$\frac{11}{32}$″	·1780	$1\frac{1}{4}$″	1·953	$1\frac{15}{16}$″	13·396	$3\frac{3}{4}$″	52·735
$\frac{3}{8}$″	·2441	$1\frac{5}{16}$″	2·261	2″	15·625	$3\frac{1}{4}$″	58·183
$\frac{13}{32}$″	·3250	$1\frac{3}{8}$″	2·599	$2\frac{1}{8}$″	18·088	4″	64·000

Where M = Torsional movement in inch lbs.

d = Diameter of shaft in inches.

f = Safe torsional stress in lbs. per sq. inch.

Then $M = 0{\cdot}196 d^3 f$ . . . . . (31)

The torsional stress may be fixed at 5,000 lbs. per square inch for carbon steel, and 7,000 lbs. per square inch for nickel and nickel chrome steel.

As an example, assuming the torsional moment to be transmitted is 1,600 inch lbs. ; then we should obtain—

$$0{\cdot}196 d^3 \times 500 = 1600$$

and $$d = \sqrt[3]{\frac{1600}{0{\cdot}196 \times 5000}} = 1{\cdot}28 \text{ inches diameter} \quad (32)$$

Hollow Shafts. In clutches it is frequently found necessary to make the clutch shaft hollow. The following is a simple method of obtaining the dimensions of a hollow shaft, which can be substituted for a solid shaft of equal strength. *See* Fig. 55A and 55B.

Let D_1 = diameter of solid shaft.

D = outside diameter of hollow shaft.

d = inside ,, ,, ,,

$t = \frac{1}{2}(D - d)$ = thickness of metal of hollow shaft.

$K = d\ D$ = ratio of diameters of hollow shaft.

As the hollow shaft is to have the same strength as the solid shaft, the moment of resistance must be equal, hence—

$$\pi(D^4 - d^4) = \frac{\pi D_1^3}{32} \text{ from which } D^3 - \frac{d^4}{D} = D_1^3$$

If KD is substituted for d in above equation

$$D^3 - D^3 K^4 = D_1^3 \text{ from which } \frac{D}{D_1} = \sqrt[3]{\frac{1}{1 - K^4}} \quad . \quad (33$$

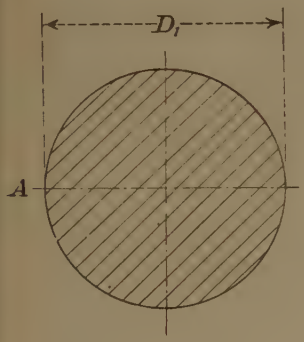


Fig. 55a.

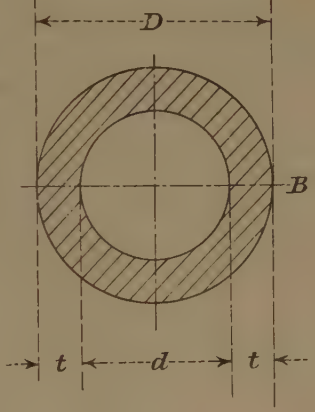


Fig. 55b.

Clutch Shaft Dimensions.

In a similar manner by substituting $\frac{d}{K}$ for D

$$\frac{d}{D_1} = K^3 \frac{1}{1 - K^4} \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

Further as $t = \frac{1}{2}(D - d)$ the following is found by substitution and simplification—

$$\frac{t}{D_1} = \frac{1 - K^3}{2} \frac{1}{1 - K^4} \quad . \quad . \quad . \quad . \quad . \quad (35)$$

Taking the previous example of solid shaft = 1·28 inches

diameter to be replaced by a hollow shaft ratio K being 0·5, then by inserting the value from equation (33).

$\frac{d}{D_1} = 1{\cdot}021$. And $D = 1{\cdot}28 \times 1{\cdot}021 = 1{\cdot}307$ inch diameter.

Square Shafts. The ends of clutch shafts are frequently designed with squared portions, and care must be taken that the section is of such proportions as will ensure the shaft's rigidity.

The fact must be borne in mind that with a given torsional moment, the square shaft, which has the same area as a round shaft, is subjected to a maximum stress 35 per cent greater than that to which the round shaft is subjected. For equal strength, therefore, the square shaft must be larger.

To find the equivalent size of a square shaft, we have already shown for a round shaft that—

$$M = 0{\cdot}196d^3f$$

denoting the side of the square by s we then obtain—

$$M = 0{\cdot}208s^3f$$

If, therefore, the square and the round shafts are to be equally stressed with a given value of M, we must have—

$$0{\cdot}208s^3 = 0{\cdot}196d^3$$

$$= \quad s^3 = d^3\frac{0{\cdot}196}{0{\cdot}208}$$

$$= \quad s = 0{\cdot}98d \qquad . \quad . \quad . \quad (36)$$

If a round section shaft be used with squared ends, these should be of such dimensions as will stand the stress to which the shaft is subjected.

It is to be regretted that this point is often overlooked, with the result that although the body of the shaft has a diameter suitable for the work, the ends are reduced to square section, the result being twisted and ultimately broken shafts.

It is recommended that holes for square shafts for all

kinds of fits be made standard size, and that the size of the shafts be varied to give the fit required.

The relation between the diameter across the corners (A), and the width of the square (B), Fig. 56, is in a ratio of 0·73 to 0·80, according to the fit required.

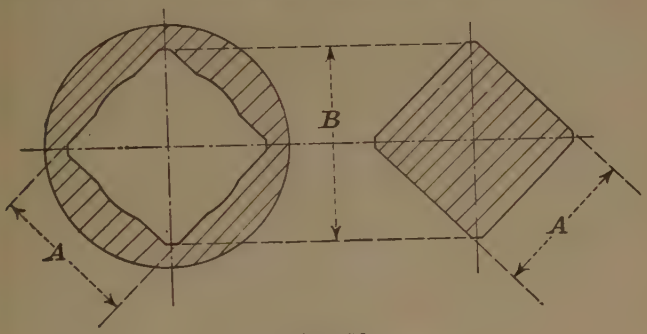


Fig. 56.
Square Clutch Shafts.

Splined Shafts. In order to keep down the weight, and at the same time provide a greater key bearing area, the integral spline shaft was developed. Such a shaft is simply a round shaft in which the splines are produced by milling out the metal between them. The splined holes through the hub of the part to be fitted can be produced accurately and cheaply by broaching or slotting. There is no doubt that this method of forming a shaft drive is superior to any other.

As now used in various classes of service, spline shafts are constructed with from 4 to 10 splines, but probably standardization of 6 and 10 splines will be found the most useful.

In Fig. 57 are shown the sections of a hub containing 6 and 10 splines respectively. The proportions of the two types shown have been used by the writer, and have been found entirely suitable for clutch shafts.

Each of these shafts are divided into three different sizes, A B C, the following formula giving the dimensions

of the various parts of the bore, while the corresponding parts of the shaft are made $\frac{1}{1000}$th of an inch on small shaft diameters, and $\frac{2}{1000}$ths of an inch less on large diameters. The widths of the splines remain constant in all cases. The other dimensions, however, have slight variations.

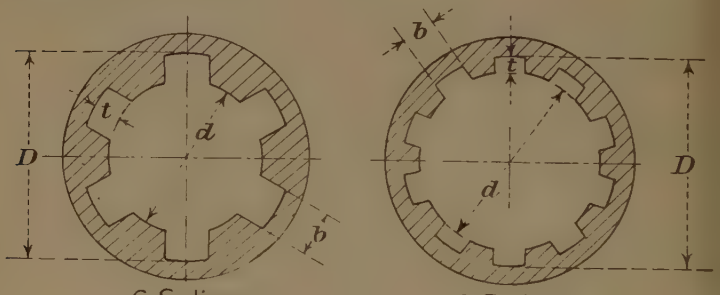


FIG. 57.

SPLINED CLUTCH SHAFTS.

A	d	= 0·90D.	A	d	0·91D.
	b	= 0·25D.		b	0·156D.
	t	0·05D.		t	= 0·045D
B	d	= 0·85D.	B	d	0·86D.
	b	= 0·25D.		b	0·156D.
	t	0·075D.		t	= 0·07D.
C	d	0·80D.	C	a	0·81D.
	b	0·25D.		b	0·156D.
	t	0·10D.		t	0·095D.

The diameter at the bottom of the splines should be calculated necessary to transmit the torque. When this is done the spline shaft will really be a stronger shaft than a round shaft of that diameter having inserted keys.

If it becomes necessary to form the splines on a shaft of certain consistent diameter, care should be taken to see that the splines merge gradually into the diameter, in order that there may be no concentration of stress at a sudden change in the section.

Keyed Shafts. It is frequently found essential to fix the driven member of the clutch permanently on the

clutch shaft by means of a taper in conjunction with a key. The object of a taper is to obtain greater friction between the hub and the shaft, the key being considerably relieved thereby.

This method slightly reduces the strength of the shaft, but just how much can only be conjectured, and is not a serious matter.

Regarding keys, it may be said that their principal function is to prevent relative motion between two parts of a mechanism.

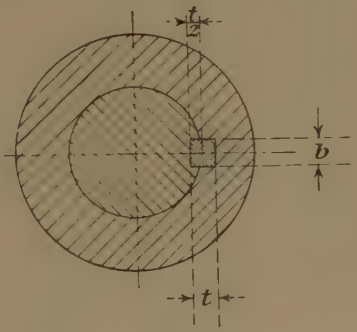


FIG. 58.
STRENGTH OF KEYS.

There are innumerable types of keys, but two kinds of sunk key will be found to cover most designs for clutch shaft purposes.

Keys are generally proportioned by formulae, based upon the shaft diameter. This is not altogether correct, as the length of the key should be taken into consideration. It will therefore be found desirable to calculate the size of key necessary, and modify these dimensions to suit practical consideration, by adopting the nearest standard size.

It is generally supposed that key failures are due to shearing, but investigations prove failures are more often the result of crushing the side of the keys.

Formulae for Keys. The crushing and shearing strengths can be calculated from the following (*see* Fig. 58) where—

T = Torsional moment.
Sc = Crushing strength.
Ss = Shearing strength.
l = Length of the key.
b = Width of the key.
t = Thickness of the key.

$$\text{Then } Sc = \frac{4T}{tld} \quad . \quad . \quad . \quad . \quad . \quad . \quad (37)$$

$$\text{and } Ss = \frac{2T}{bld} \quad . \quad . \quad . \quad . \quad . \quad . \quad (38)$$

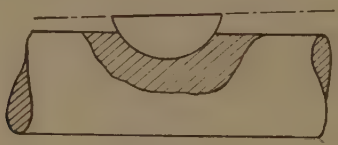


Fig. 59.
Woodruff Key.

Equating the value of T we obtain—

$$t = 2b\frac{Ss}{Sc} \quad . \quad . \quad . \quad . \quad . \quad . \quad (39)$$

If $Sc = 2Ss$, as is generally assumed, then equation (a) calls for a square key. To facilitate fitting, the width of the key is often made greater than its depth, which has the effect of decreasing Ss relative to Sc, from which it is evident that the crushing strength should form the basis of design.

Keys should always be fitted parallel to the shaft.

The Woodruff Key. In Fig. 59 is shown a modified form of sunk key, and it will be noted that the seat in the shaft has a circular outline, and is deeper than the ordinary. This weakens the shaft slightly, but the deep base precludes all possibility of tipping. The freedom of the key to adjust

itself to the seat of the hub, makes it almost impossible to obtain a bad fit, while with the ordinary taper key a good fit is difficult to obtain.

These keys are advocated for clutch shafts, and if the hubs are long, the depths of the keyways may be diminished by using two or three keys at intervals in the same key seat.

Engaging Mechanisms. Toggle mechanisms are frequently incorporated in designs of friction clutches, for

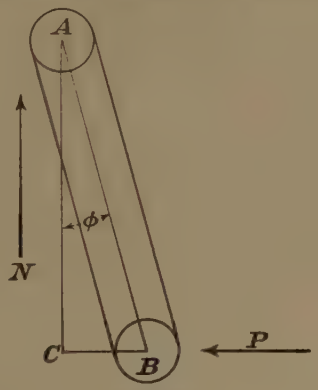


Fig. 60.

Forces in Link-engaging Mechanism.

the purpose of multiplying spring pressure at the contact surfaces.

In Fig. 20 is illustrated a clutch having such a device, and it will be noted that the multiplying the factor attains the infinite as the toggle assumes a radial position.

In practice it is essential that the toggle links can never reach the radial position, as the clutch would then have the inclination to disengage. This is prevented by adjusting the set screws accordingly.

Referring to Fig. 60, it is evident that if the links make a small angle ϕ with a radial line, then the multiplying factor is equal to contangent. AB represents diagramatically a link of the toggle, P equals the pressure of the

spring, and N the radial pressure exerted on the arm of the bell crank. Now if B be moved the slightest distance under the pressure of the spring P so that the angle BAC (ϕ) decreases to $\phi - d\phi$ then we have—

$$CB = AB \sin \phi$$
$$AC = AB \cos \phi$$

When ϕ decreases to $\phi - d\phi$, $AB \sin \phi$ decreases by $AB \cos \phi \, d\phi$, and $AB \cos \phi$ increases by $AB \sin \phi \, d\phi$, but the product of the force into the distance through which it operates represents the work done, and this will be the same at both points A and B.
Hence—

$$P \times AB \cos \phi \, d\phi = N \times AB \sin \phi \, d\phi \quad . \quad (40)$$

and $$\frac{N}{P} = \frac{\cos \phi}{\sin \phi} = \cot \phi \quad . \quad . \quad . \quad . \quad . \quad (41)$$

Referring to the multiple disc clutch shown in Fig. 20, it is evident that engagement is made by a lever arm through the medium of a mechanism which is capable of increasing the leverage rapidly toward the end of the lever displacement.

It will generally be found that graphical methods of determining the force required at the end of the leverage are the most convenient. Therefore, in Fig. 61, is shown the graphical analysis of the forces acting on the mechanism of the clutch indicated.

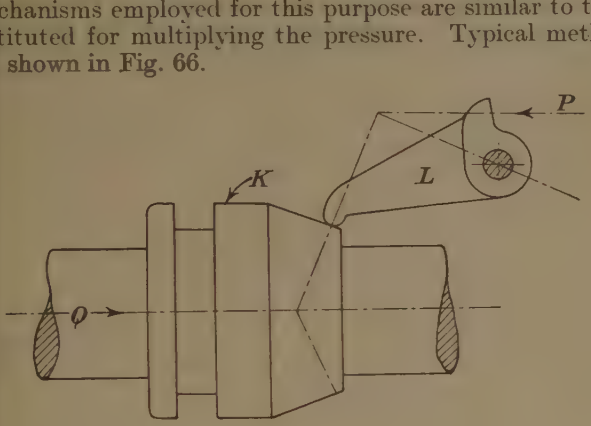
The sector AB represents the force Q exerted upon the collar K by the operating lever. Assuming that the clutch is provided with two toggle levers L, the force exerted by K upon L is represented by the sector AC. The lever L is acted upon three forces as shown in the figure. The magnitude of P is represented by CD.

The analysis of the forces acting upon the second toggle lever is given by the triangle BCE, in which the sector EC represents the magnitude of the force corresponding to P. The axial force produced is represented by the sector ED.

There are many other toggle mechanisms in general use on clutches, but these are examples in general use. Other

pressure multiplying mechanisms are described in reference to Figs. 22, 23, 25, 44.

In clutch design it is often found necessary to reduce the spring pressures at the source of operation. The mechanisms employed for this purpose are similar to those instituted for multiplying the pressure. Typical methods are shown in Fig. 66.



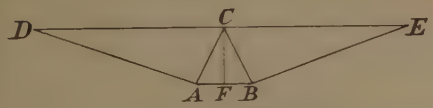


Fig. 61.

Lever Engaging Mechanism.

Operating Collars. To disengage a cone clutch the driven cone must be withdrawn from the driving cone against the exerted pressure of the clutch spring. Obviously this renders necessary the institution of a shifting mechanism, which usually consists of a sliding collar and some form of lever control.

The movable portion of the clutch is usually provided with a circumferential groove, as illustrated in Fig. 61, in which is fitted an operating collar. If both the flanges

of the groove are formed on the clutch centre, the operating collar has obviously to be made in two parts, to be made capable of assembly, and there are innumerable ways of arranging this.

When the clutch is withdrawn the entire pressure of the clutch spring comes upon the operating collar, for which reason a ball thrust bearing is usually interposed, as shown.

In the design illustrated the collar itself is made of brass, and provided with two radial pins, with which engage the

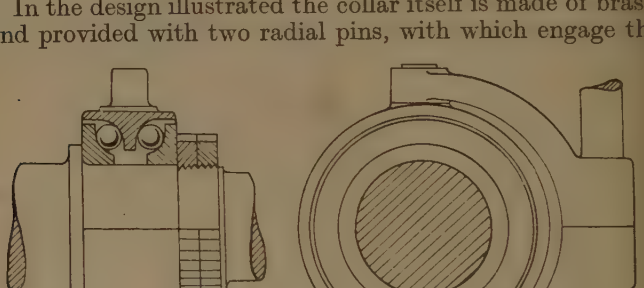


Fig. 62.

Clutch Operating Collar.

free end of the forked shifting lever, which are formed with elongated holes for the pins to pass through. These pins move in an arc of a circle, while the operating collar moves in a straight line. A greaser is usually fitted to the trunnions, for lubricating purposes.

Another common design consists in forming the operating collar as a circular disc, the operating levers being made with cam-shaped ends, working against this disc.

In automobiles pressure has to be transmitted in one direction only; consequently it is only necessary to hold the shifting fork against this collar by means of a spring, the groove being dispensed with.

Clutch Brakes. In addition to securing smooth and

easy clutch engagement, it is necessary to make provision against "spinning" of the clutch when it is disengaged. By lightening the clutch cone, "spinning" has been greatly reduced.

This is a most essential point in clutches appertaining to automobiles, while the reduction of gear shocks in changing vehicle speed is another factor of importance in reference to clutch design.

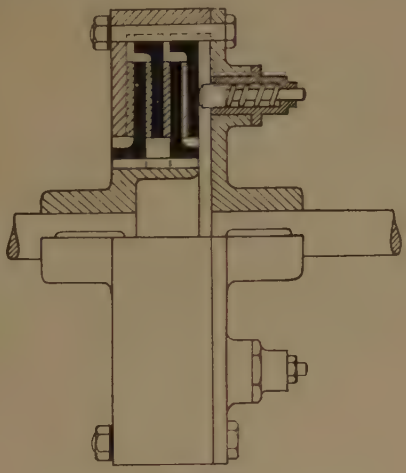


FIG. 63.

SLIPPING CLUTCH (WIGGLESWORTH).

The shocks due to gear changing are entirely produced by the unequal pitch velocities of the gears when meshing, and if the speeds of the two gears can be made to correspond, then shocks will be eliminated.

In ascending a hill it may be necessary to change to a lower gear, and obviously if the higher gear is disengaged, then the driven wheel of the lower gear will run too fast, or the driving wheel too slow, to allow of perfect meshing.

This is usually overcome by allowing partial engagement of the clutch to speed up the pinion of the lower gear.

It will be understood that as soon as the clutch is disengaged, the engine, freed of its load will accelerate its speed ; consequently if the clutch be momentarily engaged, it will also be speeded up.

Conversely in changing from a lower to a higher gear the driven gear to be meshed will be running too slowly, and the driven gear too fast to obtain perfect engagement. However, to overcome this, small brakes are frequently adopted to reduce the speed of the driven gear to a correct

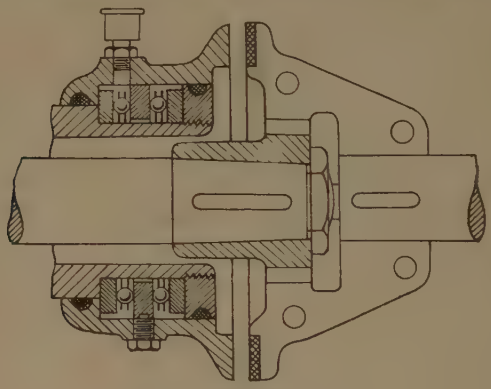


Fig. 64.

Disc Clutch Brake.

speed. Such an auxiliary brake fitted to a cone clutch is illustrated in Fig. 14.

A friction block is carried on an arm on the clutch operating lever, which comes into operation on the clutch boss when the clutch is fully withdrawn.

In the main there appears to be three designs under which clutch brakes may be classified. Firstly, that class previously shown, in which suitably placed levers forming the operating gear move a pad on to a revolving portion of the clutch.

Secondly, there follows that type in which a plain disc of fairly large diameter is brought up against a stationary ring, sometimes provided with recoil springs

to magnify its soft engagement. This type is illustrated in Fig. 64, wherein it will be seen the power is transmitted from the clutch shaft through a splined dog and coupling, which latter is attached to the transmission shaft.

The coupling is of disc form, and is provided on its face with a friction ring, which when the clutch is disengaged fully, comes into contact with a corresponding disc secured to the trunnion of the operating gear, which latter, of course, is stationary.

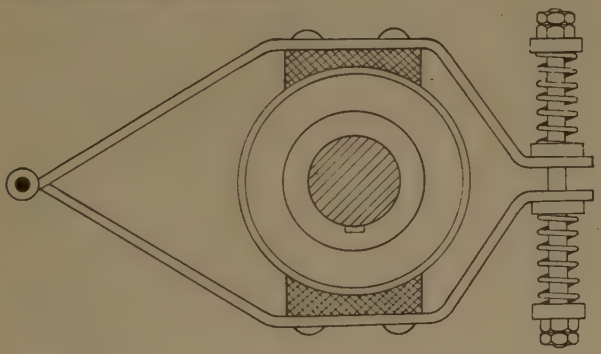


Fig. 65.

Block Clutch Brake.

The third type is that involving the use of coned or double coned surfaces engaging exactly as the ordinary cone clutch. An example of this design is illustrated in Fig. 33, but it is more usual to find the contact surfaces of steel upon fabric lining.

Where clutches are large and heavy, it is often essential to employ a more efficient method of stopping the driven member. This usually entails more complicated mechanism. An example is shown in Fig. 65, wherein it will be seen two brake blocks operate upon the revolving member. These blocks are brought into engagement by two springs, which are allowed to act by the movement of the cam, in turn operated by the clutch disengaging gear.

There are a great number of designs based on those shown, but the underlying principle of them all is to bring some part attached to the rotating clutch into contact with a part of the mechanism which does not rotate when the clutch is disengaged.

The disadvantage of clutch brakes lies in the fact that they require minute adjustment, so that the revolving member may not be stopped too quickly. Such adjustment calls for considerable care and continuous attention.

Ball Bearings for Clutches. If modern clutch designs point to one thing in particular, it is the desirability of installing ball bearings throughout.

Ball bearings are used in preference to sliding bearings for the following reasons: There is less loss of power on account of the much lower coefficient of friction; the friction of a ball bearing of the viscosity of the lubricant, or its temperature; the frictional resistance at starting is negligible, and their durability is extraordinary.

Journal Ball Bearings. This type of ball bearing is fitted to the clutch shaft at the spigot end invariably, and less frequently to the clutch cone. The reasons that influence their incorporation may be stated as—

1. Lubrication.
2. Heat generation.
3. Alignment.

Lubrication of the bearings in a clutch presents considerable difficulty, and if plain bearings are installed, such lubrication would require to be constant, exact in proportion and quality. Ball bearings excel in this respect—that they require practically no lubrication, provided water and dirt can be excluded.

More often than not clutch shaft bearings do not get efficiently lubricated, partially due to the generation of heat during clutch slippage, which evaporates any lubricant that may be in evidence. Again, a plain bearing may tend to "seize," and it then becomes impossible to free the clutch.

The clutch shaft may be fixed permanently to the

driven member, or it may be slidably mounted, but whichever method is adopted, the self-aligning type of ball bearing is strongly advocated.

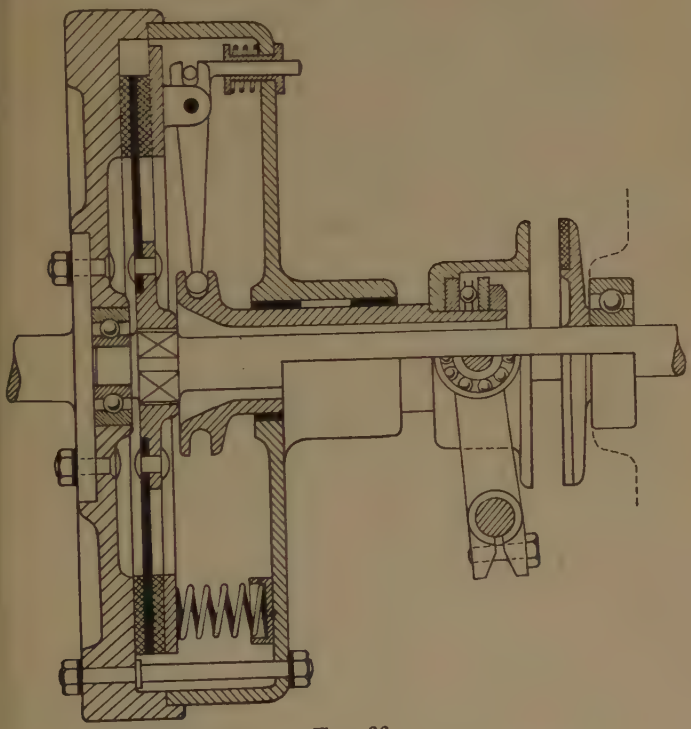


FIG. 66.
BALL-BEARING CLUTCH.

The driven member of the clutch, be it of the cone or disc type, should not be restrained in any way from coming into perfect frictional contact with the driving member, and in all cases the self-aligning ball journal bearing will enable this objective to be realized, where the shaft is inaccurately mounted, or deflects under load.

The method of application is shown in Fig. 66, which represents a composite clutch, having ball bearings in all probable positions. The normal type of ball journal bearing may be used in clutches where the driven member is able to flex independently of the shaft, as in Fig. 24 and Fig. 25, the advantage of the normal type over the self-aligning type being its slightly greater load carrying capacity, although this is not of great moment.

In the mounting of these ball bearings there are certain requirements which apply generally, although the exact arrangement is governed to some extent by conditions.

Where the bearing is to carry radial load only, the inner race should have a light drive fit on the shaft, and be securely clamped against a shoulder by a locked nut.

The outer race of a radial bearing should fit closely in its housing, but be free to float or move in an axial direction. Fitting in this manner will ensure correct distribution of the load over the outer race.

Another feature in favour of these bearings is that fitting is an easy operation, and entirely eliminates scraping and bedding in by hand.

In Fig. 66 will be seen a method of adapting ball journal bearings in the clutch operating gear. The ball bearings are mounted on pins fixed to the operating arms. In pressing forward the arm to declutch, the spinning of the end of the driven member revolves the bearings on the pins, and thereby reduces the friction induced by the pressure of the clutch spring.

In line-shaft transmission the clutch shaft is usually a long shaft. The use of ball journal bearings of the self-aligning type cannot be too strongly recommended for this purpose, as apart from their power-saving advantages, they are capable of taking all normal thrusts.

The type suggested is shown in Fig. 67.

Ball Thrust Bearings. These bearings are adapted for heavy axial loads. The best form has two grooved races of circular section, with flat seats, or preferably one spherical seat, so that the latter will align itself, in order to distribute evenly over the entire number of balls.

When mounting thrust bearings of the single type,

one ring should be a tight fit on the shaft and rotate with it, and the other ring should remain stationary, except in the case of the spherical seat, which should have a certain

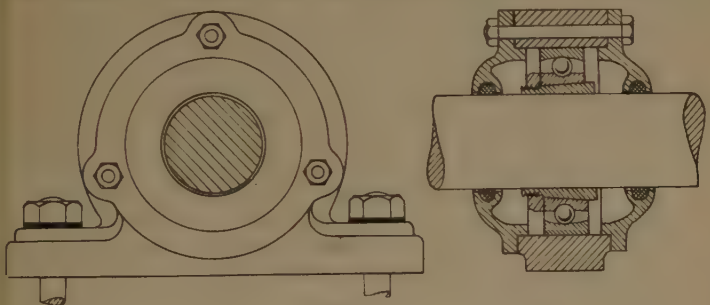


Fig. 67.

Line Shaft Transmission Bearing (S.R.O.).

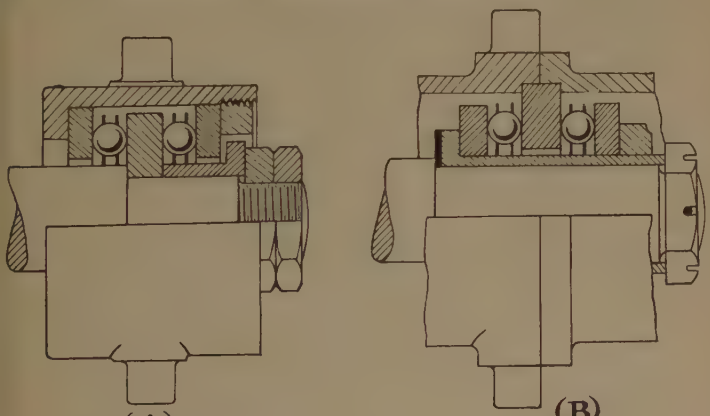


Fig. 68A. Fig. 68B.

Mounting Double Thrust Bearings.

latitude for adjustment, to compensate for changes in shaft alignment.

In double thrust bearings the centre ring should be definitely located as in Figs. 68A and 68B.

In clutches ball bearings are used to the spring thrust when the clutch is engaged, and also the pressure of the operating mechanism to overcome the spring force when declutching. When the clutch is disengaged, and at rest, its spring bears with one end against a rotating part, and the other against a rotating member, and to prevent undue friction and wear a ball thrust bearing is introduced.

If no ball thrust bearing were provided, the friction between the spring and its support would probably be great enough to cause the clutch to keep spinning when disengaged.

The ball thrust bearing is usually mounted by passing over the end of the shaft, and held in place by screwed and locked nut or cap. With either arrangement the spring pressure should be adjustable.

Methods of installation of these bearings are shown in Figs. 11, 13, 14, 15, 16, 51, 52, 53.

CHAPTER IX

MACHINE TOOL CLUTCHES

USUALLY the governing factor in clutches for this purpose consists in arranging the clutch in a compact manner, as the available space is usually limited. Fortunately the powers to be transmitted are only moderate.

Split Ring Clutches. The split ring type of clutch is particularly suitable for machine tools. In Fig. 69 is shown the well known clutch made by the Carlyle Johnson Machine Co., Manchester, Connecticut, U.S.A., and it will be seen that this clutch consists of an outer shell running loose on a shaft, into which is fitted a split ring.

This latter is expanded by the action of a pair of levers, which in turn are expanded by the action of an operating collar with a wedge. The combined action engages the split ring with the outer shell, and forms a most efficient clutch for lathes and drilling machines.

Formulae for Split Ring Clutch. It is assumed that the pressure exerted by the split ring on the clutch is uniformly distributed over the area in contact, hence the expression for the frictional moment is as follows, where—

M = Torsional moment.
μ = Coefficient of friction.
D = Diameter of split ring.
f = face of split ring.
P = normal unit pressure.
θ = Angle of contact (assumed 360°).

Then $dM = \dfrac{upfD^2d\theta}{4}$

and $M = \dfrac{u\mu PfD^2}{2}$ . . . . . (42)

The horse power transmitted by this clutch at N revolutions per minute is—

$$HP = \frac{\mu\pi^2 PfD^2 N}{396000} \quad . \quad . \quad . \quad . \quad . \quad (43)$$

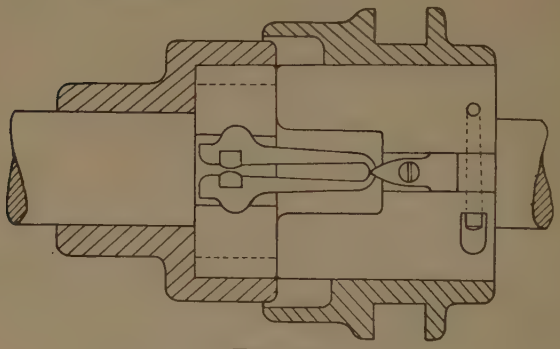


Fig. 69.
Split-ring Clutch (Johnson).

The shell of the clutch is made about $\frac{1}{64}$ inch larger on its internal diameter than the external diameter of the split ring; consequently a certain part of the force P exerted by the operating mechanism is used in pressing the ring outward. Again, when the ring comes into frictional contact, a further force is required to press the ring against the shell.

Denoting these two forces by P_1 and P_2, their magnitude must obviously be equal to P.

The formulae for these forces has been calculated as follows—

$$P_1 = \frac{E1}{2r^1}\left(\frac{1}{r^2} - \frac{1}{r^1}\right)$$

where P^1 and P^2 represent the initial and eventual radii of the split ring respectively.

The pressure upon an elementary length of the ring is—

$$\frac{PfDd\theta}{2} \quad . \quad . \quad . \quad . \quad . \quad (44)$$

and the moment of this pressure about section at A is

$$dM_A = \frac{PfD^2 \sin \times \theta d\theta}{4}$$

from which we find that the bending moment at that section of the ring is

$$M_A = \frac{PfD^2}{2} \quad . \quad . \quad . \quad . \quad (45)$$

Since this bending moment must equal the force P_2, it is obvious that—

$$P_2 D = \frac{PfD^2}{2}$$

and

$$P_2 = \frac{PfD}{2}$$

from which we obtain—

$$P_2 = \frac{M}{\mu \pi D} \quad . \quad . \quad . \quad . \quad (46)$$

Machine Tool Cone Clutch. An excellent example of the application of the cone clutch to machine tools is illustrated in Fig. 70, wherein it will be seen that the driving pulley runs loose on a bearing, and has an external cone attached to it. The clutch cone is fixed to the main driving shaft on a sliding key, and is thrust into engagement by a tapered operating collar acting upon the toggle mechanism described in Fig. 61. This clutch is designed on the principles previously outlined, and is extremely simple and effective.

Machine Tool Block Clutch. This type of clutch can be constructed in so small a compass, that it is eminently suitable for this class of work, and is in common use for reversing feed mechanisms.

In Fig. 71 is shown this type of clutch, wherein it will be seen that a pair of bevel gears run loose upon the shaft, into which are fitted clutch shoes, loosely connected to a slidably mounted sleeve.

These shoes are pressed against the inner surface of the drum formed inside the bevels by the sliding sleeve,

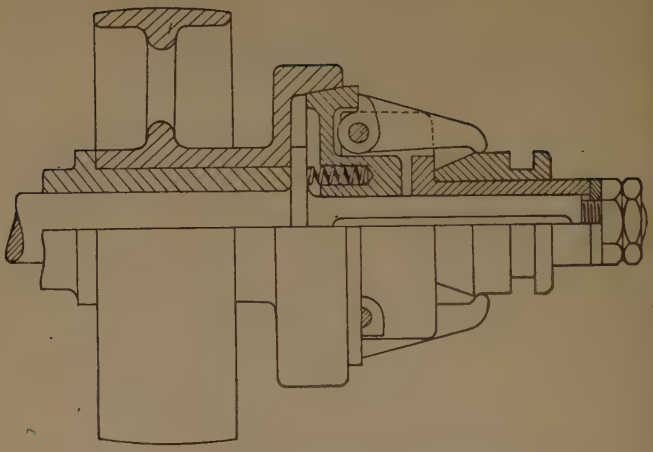


Fig. 70.
Machine Tool Cone Clutch.

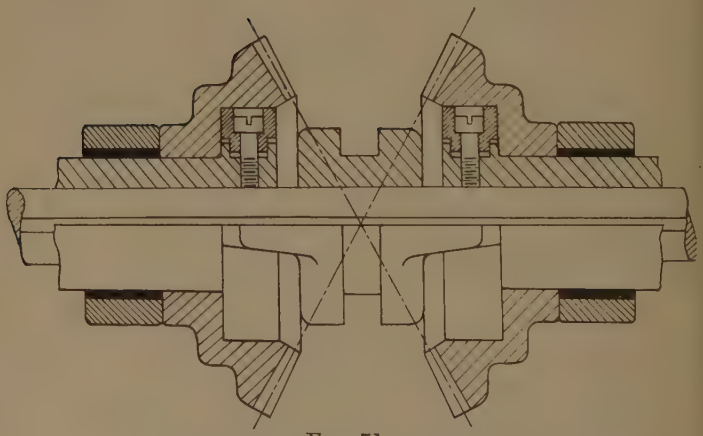


Fig. 71.
Machine Tool Block Clutch.

which is formed integral with the double wedges, which are made capable of sliding along the inclined surfaces of the shoes. The shoes are connected to the sleeves by set screws, and therefore rotate with them.

The shaft may be either the driving or the driven member of the clutch, and it is evident that this method enables quick reversal of the mechanism to be performed.

The calculations for this clutch are, of course, exactly as previously described in dealing with block clutches.

CHAPTER X

MOTOR CYCLE CLUTCHES

CLUTCHES for motor cycles are worthy of note by reason of the exacting conditions they are required to satisfy. The space available for their incorporation is usually very scanty, and as the powers to be transmitted are considerable, clutch design must be the subject of careful study and ample experience.

Practically every type of clutch has been tried on motor cycles, and generally speaking it may be said that lack of space has condemned the use of the cone type, weight of revolving parts has designated band and block types as unsuitable, and the position has resolved itself into one eminently desiring the use of disc clutches, wherein owing to the action of multi-gripping surfaces, clutches may be instituted in confined areas.

Plate Clutches. This type of clutch is not often found on motor cycle power units ; still they may be successfully employed where the power to be transmitted is low.

In Fig. 72 is illustrated one of this type fitted to the German Magnet machine, and is embodied in the belt pulley on a countershaft. The two plates inside the pulley are pressed apart by eight springs, and are connected to the hub carrying the small chain wheel driving to the rear wheel.

In the end of the shaft, which does not revolve, is a quick screw thread, in which operates a lever pressing against a collar when the lever is turned in a right hand direction, thereby drawing the two driving members together, and consequently releasing the clutch.

The good points of this clutch consist in the equalization of pressures when the clutch is either engaged or declutched, and the ease of manipulation obtained by means of the screwed lever.

Multiple Disc Clutches. Practically all kinds of multiple disc clutches have been used for motor cycle purposes.

The V groove type has not been very successful, owing to its inherent disadvantage of slow disengagement, due to the oil between the plates, and the difficulty in dissipating the heat generated by considerable slippage.

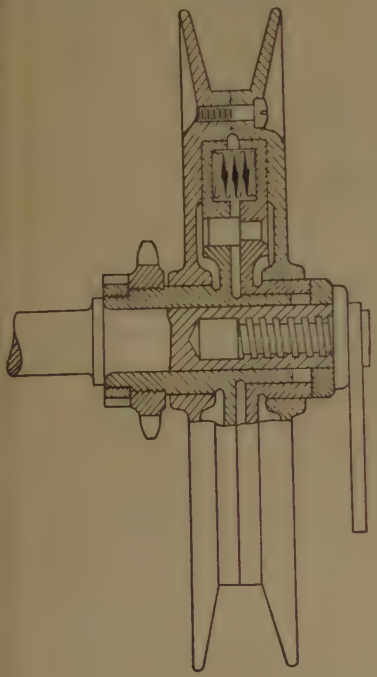


FIG. 72.

MOTOR-CYCLE CLUTCH (MAGNET).

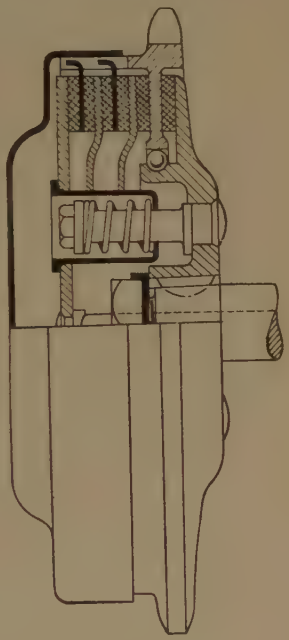


FIG. 73.

MOTOR-CYCLE MULTIPLE DISC CLUTCH.

The ability to slip for considerable periods is a very necessary factor for this class of work, and is probably the controlling factor in design.

Hence it has been found that metal to metal clutches are not very suitable, and multiple discs having alternate

discs of fabric, or inserts of fabric or cork, now represent standard practice.

In Fig. 73 is shown a clutch which has six discs of "Ferodo" material operating between steel surfaces. The chain wheel is driven from the engine, and runs on a ball bearing whose inner race is formed on the plate fixed

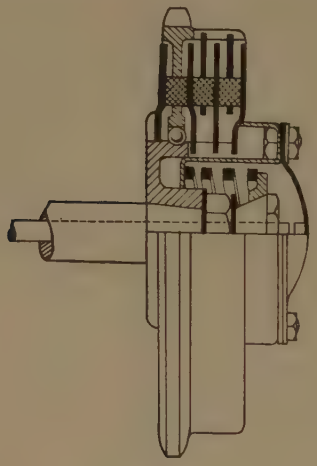


Fig. 74.

Motor-cycle Clutch (Cork Inserts).

to the gear shaft. Keyed slidably to the chain wheel are two steel discs, while two driven discs are connected in a similar manner to the pressure plate.

The friction surfaces are kept together by the pressure springs working on pins suitably connected to the driving plate. Withdrawal is effected by a quick thread screw and lever attached by wire to a lever mounted on the handlebars of the machine.

Multiple Disc Insert Clutch. This clutch, which is a very successful type, is similar in principle to the preceding

example. A design is shown in Fig. 74, as used on a well-known motor cycle.

It will be noted that circular fabric inserts are used in this design, the advantages and disadvantages of which have been previously enumerated, while a single clutch spring is used, the whole forming a most simple and compact arrangement.

CHAPTER XI

MISCELLANEOUS

THE following chapter is devoted to particulars of special devices lately ascertained, and of interesting features of other clutches, or points of design omitted from previous descriptions.

Roller Clutch. In a previous chapter it was mentioned that roller clutches were classified as a type of rim clutch.

As a matter of fact, clutches of this type are not friction clutches in the generally accepted sense, nevertheless they present some interesting features, and are considerably used in certain classes of machinery.

In Fig. 75 is illustrated a design used for punching press. In this it will be seen there is a cam keyed to the crankshaft, which has a series of inclined planes cut upon it in the form of recesses. The rim in this instance forms the hub of the flywheel, between which and the cam is interposed a cage containing rollers. These rollers are forced up the inclined planes by the action of the cage, and thereby wedge themselves between the cam and the rim, making a definite connection between the crankshaft and the flywheel.

The rollers are held in position by the cage, which is connected to the crankshaft by a spring.

In working, the roller cage is connected by a lever to a pedal which, when released, allows the spring to rotate the cage a little way around the shaft, taking the rollers with it. This action causes the rollers to wedge between the cam and the rim, a rigid connection being thereby established, the flywheel and crankshaft rotating together.

Disengagement is arranged by depressing the pedal, thereby forcing the cage and rollers back into their original position, permitting free rotation of the flywheel again.

Waring-Brown Shock Absorber. This is an adaptation of the roller clutch used in conjunction with a small cone clutch, patented by the author for use on automobiles, as a

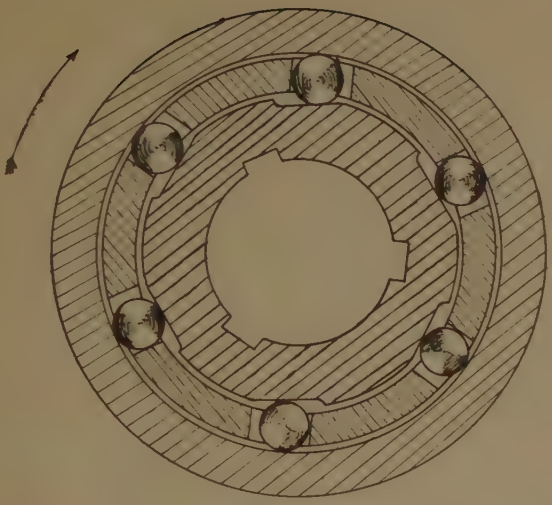


FIG. 75—PUNCHING PRESS CLUTCH.

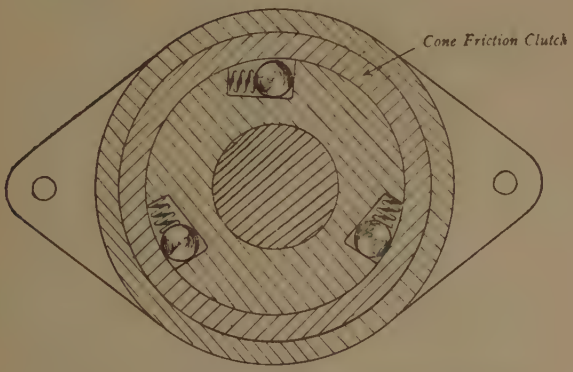


FIG. 76—SHOCK ABSORBER CLUTCH (WARING-BROWN.)

shock absorbing device acting between the road wheels and the chassis.

It is a well known theory that resistance to the action of the chassis springs should be in evidence in one direction only, which is the principle forming the basis of this design in Fig. 76. The outer casing is definitely fixed to the frame of the vehicle, and is formed internally as the outer cone of a friction clutch.

The internal cone is pressed into frictional contact by a spring. A shaft runs through the centre of the device and has formed upon it three inclined planes. Fixed to the end of the shaft is a lever which is connected by a link to the axle carrying the road wheels.

The operation in this instance is as follows : When the automobile is running, and the wheels come into contact with a bump, the axle is forced in an upward direction and with it the link connected to the lever of the shock absorber. In the partial rotation of the shaft, the inclined planes formed on the centre simply carry the rollers round freely. When the vehicle wheel descends to its original position, it is prevented from so doing rapidly by the slight rotation of the shaft and cam of this device which owing to the inclined planes wedging the rollers on the rim of the internal cone, which is then rotated at a slow pre-determined rate due to the friction between the cone clutch. The cone is lined with Ferodo material, the rollers are kept up to their work by springs, and the operation is positive at any movement of the lever.

The Benn Friction Clutch. This is a type of plate clutch eminently suitable and widely used for industrial purposes where large powers are to be transmitted. The outstanding features are its simple and durable construction.

It will be seen from Fig. 77 that the friction surfaces are metal to metal, having an automatic lubricating arrangement for the prevention of heating and wear.

A very good feature of this clutch is that the frictional pressure between the rings and the drum is not applied direct through the striking gear as is frequently the case, but by two helical springs which are so arranged that they cannot be damaged by overload. These springs are

carried on an adjustable hook bolt, and by simply increasing or decreasing the tension, the clutch can be set to slip at any pre-determined load.

The hook bolts are attached to the ends of levers, the opposite ends of which are connected to link mechanisms for multiplication of the spring pressure. The levers are pivoted on a muff operated by the striking gear.

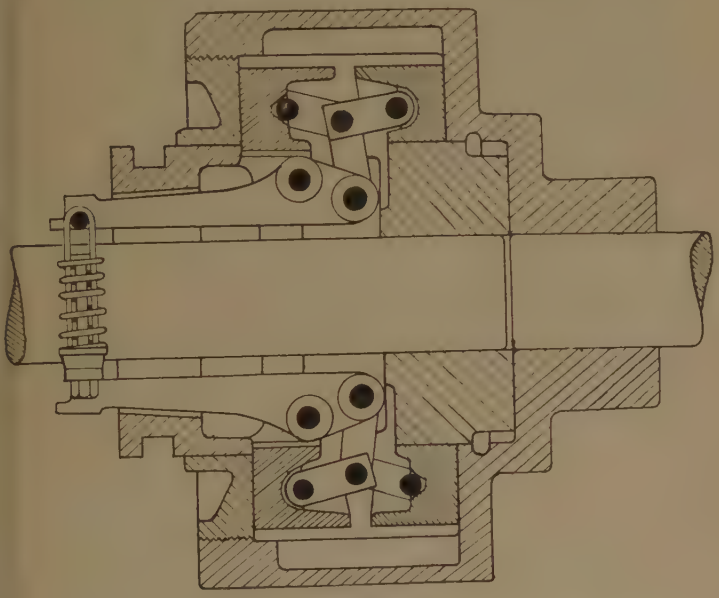


FIG. 77

"BENN" FRICTION CLUTCH (Thos. Broadbent & Sons, Ltd.)

The pushing in of the muff by the striking gear compresses the springs until the radial link passes the vertical, after which the springs, acting through the levers, automatically draw in the muff until it butts against the driven plate, the clutch then being in its engaged and locked position, where it can only be disengaged by operating the striking gear.

The counterplate is adjustable in the drum for taking up wear and is fixed in position by a set screw which engages in notches formed in the circumference of the counterplate.

Adjustment is exceedingly simple and can be made in a few minutes without dismantling any part of the clutch or striking gear.

It will be noted that the clutch complies with the requirements of a perfect device as previously enumerated, in that it is effective in either direction of rotation, centrifugal force does not affect it, while it is symmetrical in construction and perfectly balanced.

Standardization. A long experience of clutch design in all its branches has impressed upon the writer the need for standardization of details, and his attention has been given for some time past in attempts to accomplish something in this direction.

It is obviously impossible to adequately deal with the subject in a work of this kind, but it is hoped to publish later a work covering this point.

However, it is sufficient to indicate that the chief points in such standardization are those concerning—

1. The angle of clutch cone.
2. The size of clutch cone.
3. The size of clutch discs.
4. The size of clutch inserts.

The angle of clutch cone is definitely advocated as $12\frac{1}{2}$ degrees, which then only leaves the question of standardization in regard to clutch cone linings.

The diameters should be arranged in definite stages of advancement, the width being a function of such diameter. Similarly in regard to disc clutches, the friction surfaces should be arranged in definite diameters, the internal diameter being a function of the outer.

The standardization of inserts offers no difficulty whatever. In regard to thicknesses, these also appear to be easy of solution.

Nevertheless, no arrangement had been made up to the present for standardization in these respects, but it will be very obvious that such would realize a considerable saving

in time, in obtaining and fitting replacements, and would result in more economical production on the part of the makers.

To the user also such standardization would be a great boon, and particularly does this apply in reference to automobile and transport users.

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